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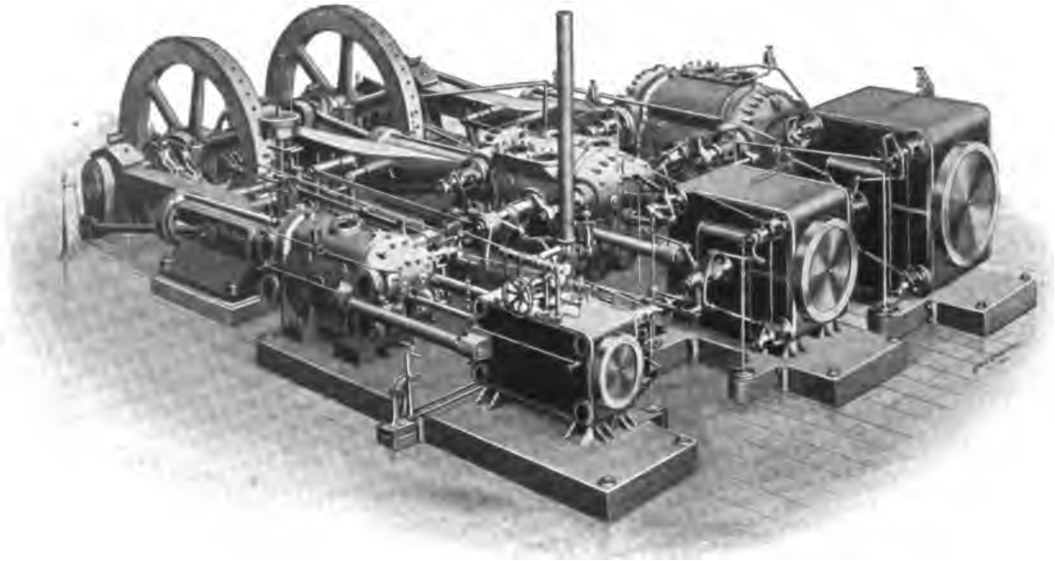

GENERAL VIEW OF THREE-STAGE TRIPLE EXPANSION AIR COMPRESSOR.
Nordberg Manufacturing Co.

# Compressed Air

*A Reference Work on*

THE PRODUCTION, TRANSMISSION, AND APPLICATION OF COMPRESSED AIR; THE SELECTION, OPERATION, AND MAINTENANCE OF COMPRESSED AIR MACHINERY; AND THE DESIGN OF AIR POWER PLANTS

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ILLUSTRATED

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1909

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# Foreword

COMPRESSED air and electricity to-day share between them the great field of power transmission, to the practical exclusion of all other transmission agents. Steam, because of its rapid condensation, can hardly be considered as a practical power transmitter for any but the shortest distances. Hydraulic transmission has had no general application. Mechanical means of transmission, such as shafts, belts, ropes, chains, and gears, are necessarily very limited in their possibilities. Compressed air and electricity, on the contrary, offer almost unlimited possibilities of extension and of diversified application; and in all cases, where properly applied, they are entirely successful when measured by the modern standards of commercial economy.

¶ With the relative merits or shortcomings of compressed air and electricity as power-transmitting agents, the pages following have nothing to do. Modern engineering has defined, with reasonable clearness, the respective fields of these agents, into either of which the other agent cannot hope to enter with any degree of success. The rivalry of the earlier days between these two forms of power transmission has given place to a mutual recognition of one another's merits and cordial coöperation in certain classes of work for which neither alone is fully adapted.

¶ Instances of this union are found: In the electro-pneumatic switch and signal systems for railway work—in which electric power moves the delicate valves controlling the application of the energy of compressed air; in the air-brake systems for street and interurban railway systems — where electrically-driven compressors furnish the air for the brake cylinders; in the very recent development of the "electric-air" rock-drill, stone-channeler, and coal-cutter—in which a pulsator driven by an electric motor produces, in a volume of compressed air, impulses which operate the cutting engine; and in the superb direct-connected electric-driven air-compressors of to-day — which have established new standards of power-driven compressor economy.

¶ Developments of recent years have been in the direction of higher economies in the production and use of compressed air; and while the ultimate limit has not yet been reached along this line, still, the compressed air appliances of to-day are highly efficient machines, measuring up to the best standards of economy. Compressed air is not "in its infancy"; rather it has attained its full majority and now awaits only a wider understanding of its value and of its possibilities in almost every phase of industrial activity.

¶ The pages which follow have been written for the practical man, for the man less interested in the finer points of theory and mechanical design than in the results accomplished and the way they are secured by the most successful builders of compressed-air machinery. No attempt has been made to draw comparisons; but the facts have been stated with the idea that the reader may form his own conclusions on all points where a discussion is possible.

LUCIUS IRVING WIGHTMAN

NEW YORK, 1909.

# Table of Contents

PRODUCTION OF COMPRESSED AIR . . . . . . . . Page 1

General Definitions — Composition and Properties of Air — Boyle's Law — Isothermal Compression — Charles's Law — Adiabatic Compression — Pressures and Volumes in Compression — Heating Effect of Compression — Compressed Air Indicator Card — Relation of Pressures, Volume, and Temperature — Power Required to Compress Air — Air-Compression at Altitudes — Unavoidable Losses in Air-Compression — Net Efficiency of Air-Compression

CLASSIFICATION OF COMPRESSOR TYPES . . . . . . Page 32

Steam-Driven and Power-Driven Compressors — Vertical and Horizontal Compressors — Straight-Line Compressors — Duplex Compressors — Single-Stage and Multiple-Stage Compressors — Compound Compressors — Other Classifications — Compound Steam Cylinders on Air-Compressors — Steam Pressure for Compounding — Regulation of Steam-Driven Compressors — Governors — Unloaders — Controllers — Starting Unloaders

AIR-COMPRESSOR PRACTICE . . . . . . . . . Page 79

Water Jackets — Compound Air-Compressors — Limiting Pressures — The Intercooler — Air-Inlet Valves — Areas of Valve Ports and Passages — Inlet and Discharge Port Areas — Air-Discharge Valves — Mechanical Efficiency of Air-Compressors — Ignition and Explosions — High-Pressure Compressors — Gas-Compressors — Design of the Air Power-Plant

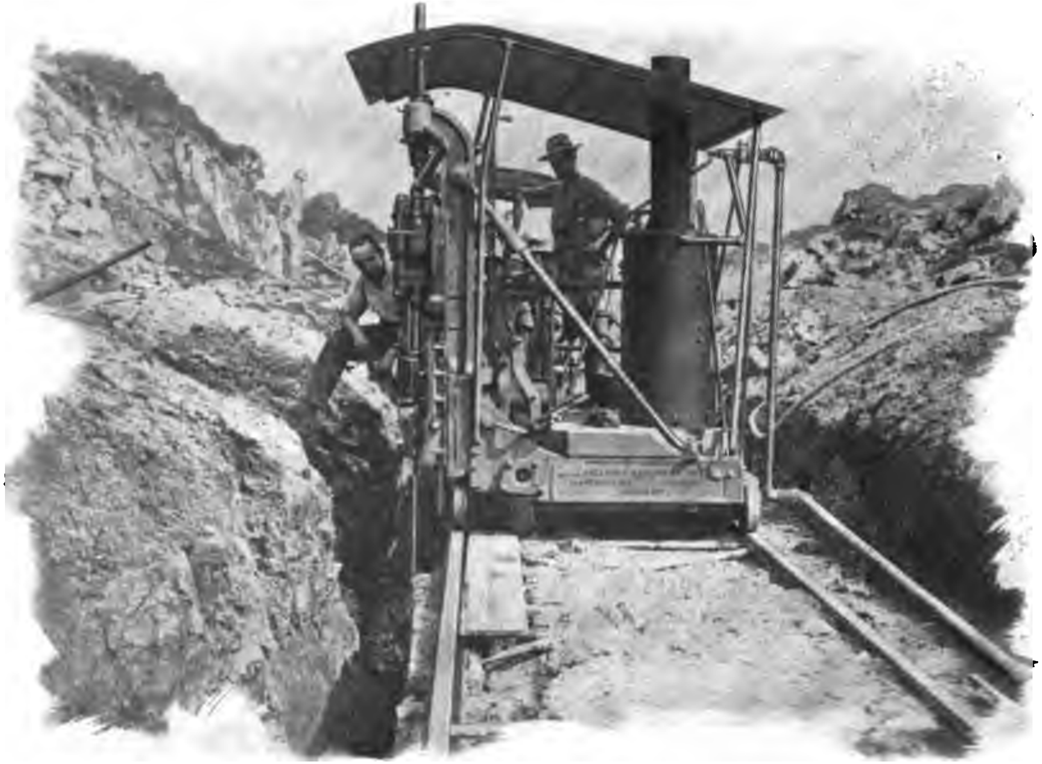
TRANSMISSION AND DISTRIBUTION OF COMPRESSED AIR . . Page 109

Tables of Air-Power Transmission — Effect of Valves, Tees, and Elbows — Necessity for Dry Air — The Anticooler — Removal of Moisture During Compression — The Aftercooler — Primary Air-Receiver — Secondary Receivers — Pipe-Line — Loop System — Dense-Air System — Reheating Compressed Air

APPLICATION OF COMPRESSED AIR . . . . . . . . Page 129

Elementary Theory — Air-Motor Indicator Card — Mean Effective Pressure for Air-Expansion — Temperatures of Expansion — Use of Compressed Air without Expansion — The Rock-Drill — Hammer Drills — Coal-Mining Machines — Track Channelers — Bar Channelers — Pneumatic Hammers — Compression Riveters — Pneumatic Punches — Pneumatic Hoists — Sand Blasts — Pumping by Compressed Air — The Electric-Air Principle — Electro-Pneumatic Switch and Signal System - Pneumatic Haulage — Bibliography

INDEX . . . . . . . . . . . . . . Page 163



COMPRESSED AIR ON THE PANAMA CANAL
Sullivan Air Channeler at Bas Obispo

# COMPRESSED AIR

## PART I

Engineering practice recognizes three primary subdivisions of any power system. The first of these is the *power producing* or *generating* element, which is always a converter of energy from some present form into another form more convenient for transmission and application to some specific purpose in hand. The second subdivision covers the *power transmitting* or *distributing* element, by means of which the newly converted energy is delivered to its point of application. The third division is the *power applying* or *utilizing* element, which is also a power converter, the delivered power being transformed into some form of useful work.

These logical subdivisions will be used in the present treatment of the subject of Compressed Air, which will be considered under the three general sections of *Production*, *Transmission*, and *Application*.

In the first element of a compressed-air power system, which consists of the compressor, the power of steam, electricity, or other energy is converted into the energy of compressed air. In the transmission system, this compressed air is carried to a point where it is to be used. In the third division, the compressed air is applied in mechanical devices, by means of which its power is converted into such work as driving machinery, drilling rock, channeling stone, chipping metal, driving rivets, pumping, operating switch and signal systems, and the other well-known applications of compressed air.

## PRODUCTION OF COMPRESSED AIR

**General Definitions.** *Free air* is air at ordinary atmospheric pressure and temperature, whatever these may be. It shows no pressure as registered by the gauge, but has an absolute pressure due to its own weight. Thus free air is in reality compressed air unconfined within any limits. Throughout this treatise, "free air," unless

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otherwise specified, will be considered as air at an absolute pressure of 14.7 lbs. and at a temperature of 32° F.

*Atmospheric pressure* is the pressure in the free air due to its own weight, or the weight of all the air above it over a unit-area. Atmospheric pressure is variable, depending upon altitude or height above sea-level and upon temperature. At sea-level and at a temperature of 32° F., atmospheric pressure is 14.6963 lbs. per square inch, or, as usually written, 14.7 lbs. Atmospheric pressure grows less as altitudes increase, and increases as depths below sea-level increase.

An *atmosphere* is an arbitrary unit of pressure representing 14.7 lbs. by the gauge. Thus "a pressure of two atmospheres" means a pressure of 29.4 lbs., gauge. Free air always has one atmosphere of absolute pressure, which must always be taken into account in computations involving absolute pressure. "Two atmospheres" gauge pressure is thus 29.4 lbs. gauge, or 44.1 lbs. absolute pressure.

*Absolute pressure* is the pressure registered by the gauge, *plus* the atmospheric pressure. Absolute pressure is therefore just as variable a quantity as atmospheric pressure, and is affected by the same conditions. For all practical purposes, however, it is usually based on the pressure of air at sea-level and at 32° F., or 14.7 lbs. Thus any absolute pressure is found by adding 14.7 lbs. to the gauge pressure.

*Absolute temperature* is the temperature as indicated by the Fahrenheit scale + 461°. Thus, at 80° F., the absolute temperature is 80 + 461, or 541° absolute; and at −20° F., the absolute temperature is −20 + 461, or 441° absolute. On the Centigrade scale, the value 461 is to be replaced by 273. Thus 40° C. represents an absolute temperature of 40 + 273, or 313° C. absolute. The Fahrenheit scale, however, will be used throughout the present discussion.

**Composition of Air.** Air is a mechanical mixture composed principally of two gases—*oxygen* and *nitrogen.* The proportions of the two by weight are 23 parts of oxygen to 77 parts of nitrogen. The proportions by volume are 20.7 parts of oxygen to 79.3 parts of nitrogen. By *mechanical mixture* is meant that these two gases are not chemically united in air, as are the oxygen and hydrogen which combine in water with the symbol $H_2O$. Air has no chemical symbol.

Air is never found absolutely pure. It may contain other gaseous substances in varying and minute quantities, among which the most common are carbonic acid gas and vapor of water. The relative proportions of oxygen and nitrogen are also known to vary within narrow limits under different conditions. The proportion of vapor of water contained in air varies with the temperature and with the pressure or density of the air.

**Some Physical Properties of Air.** A *perfect gas* is one which cannot be liquefied. Repeated experiments have practically demonstrated the fact that all gases can be reduced to liquid form under suitable conditions of pressure and temperature, and it may therefore be said that there is no perfect gas; but since nearly all gases are liquefied only with great difficulty and under the greatest extremes of temperature and pressure, most gases may for ordinary purposes be considered as perfect gases.

The gases composing air are in the latter class; and air itself, therefore, for all practical purposes, may be looked upon as a perfect gas. In fact, the behavior of air is found in practice to conform so closely with the laws of perfect gases that the discrepancies may be ignored in ordinary calculations such as those involved in the present treatment of the subject.

Table I, from Richards' "Compressed Air," lists the weight of air per cubic foot at various temperatures Fahrenheit; also the volume of one pound of air at the same temperatures.

The weight of a cubic foot of dry air at any atmospheric pressure and at any temperature Fahrenheit, may be found by dividing the constant 39.819 by the absolute temperature (temperature Fahrenheit + 461.) The volume of one pound of dry air at atmospheric pressure and any temperature, is found by dividing the absolute temperature by the constant 39.819. Thus,

Let $T$ = Temperature, Fahrenheit.
$V$ = Volume, in cubic feet.
$W$ = Weight, in pounds.

Then,

$$W = 39.819 \div (T + 461);$$
$$V = (T + 461) \div 39.819.$$

Where the temperature and pressure of the air are both variable quantities, the weight per cubic foot is found by multiplying the con-

## TABLE I

## Weights and Volumes of Dry Air at Atmospheric Pressure and at Various Temperatures

(Richards)

| Temperature, Degrees Fahrenheit | Weight of One Cubic Foot in Pounds | Volume of One Pound in Cubic Feet |
|---|---|---|
| 0 | .0863 | 11.582 |
| 10 | .0845 | 11.834 |
| 20 | .0827 | 12.084 |
| 30 | .0811 | 12.336 |
| 32 | .0807 | 12.387 |
| 40 | .0794 | 12.587 |
| 50 | .0779 | 12.838 |
| 60 | .0764 | 13.089 |
| 70 | .0750 | 13.340 |
| 80 | .0736 | 13.592 |
| 90 | .0722 | 13.843 |
| 100 | .0710 | 14.094 |
| 110 | .0697 | 14.345 |
| 120 | .0685 | 14.596 |
| 130 | .0674 | 14.847 |
| 140 | .0662 | 15.098 |
| 150 | .0651 | 15.350 |
| 160 | .0641 | 15.601 |
| 170 | .0631 | 15.852 |
| 180 | .0621 | 16.103 |
| 190 | .0612 | 16.354 |
| 200 | .0602 | 16.605 |
| 210 | .0593 | 16.856 |
| 220 | .0591 | 16.907 |

stant 2.7093 by the absolute pressure in pounds, and dividing by the absolute temperature. Under these conditions, the volume per pound is found by dividing the absolute temperature by the product of the absolute pressure and the constant 2.7093. Thus, taking the same symbols as above, with the addition of $P$ = pressure in pounds, gauge

$$W = 2.7093 \times (P + \text{atmosphere}) \div (T + 461);$$
$$V = (T + 461) \div \left\{(P + \text{atmosphere}) \times 2.7093\right\}$$

**Boyle's Law.** The First Law of Gases is generally known as *Boyle's Law*, and expresses the relation between pressure and volume in a perfect gas. This law states that, *at a constant temperature, the volume of a gas varies inversely as its pressure*; or, *at a constant temperature, the product of the pressure and volume of a gas is a constant.*

Let $P$ = Pounds pressure per square inch, absolute;
$V$ = Volume in cubic feet.

Then,

$$PV = C.$$

This is the algebraic expression of Boyle's law. The value of the constant $C$ is not the same for all gases, and indeed varies for the same gas under different temperatures.

One pound of air at 32° F. and atmospheric pressure, occupies 12.387 cubic feet. In this case, $P = 14.7$ lbs. absolute; and $V = 12.387$ cubic feet. The constant $C$, therefore, for air at 32° F. and atmospheric pressure, is $14.7 \times 12.387$, or 182.08.

**Isothermal Compression.** Fig. 1 is a graphic representation of the relations expressed by Boyle's law. Pressures are laid off on the vertical line $OY$, and volumes on the horizontal line $OX$. Any vertical line representing pressure will, with a horizontal line representing volume, enclose a rectangle, the area of which will be $P \times V$. According to Boyle's law, this product, and therefore this area, must always be a constant, and with air at 32° F. must be 182.08. If a series of such rectangles be laid out, a line joining their upper right-hand corners becomes a curve mathematically known as a *rectangular hyperbola.* In thermodynamics, this curve is known as the *Isothermal Curve,* or curve of compression at constant temperatures.

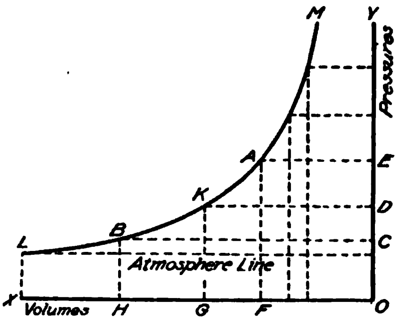


Fig. 1. Isothermal Curve Illustrating Boyle's Law of Pressures and Volumes.

*Isothermal compression* of air is compression without any change in the temperature of the air volume during compression.

**Charles's Law.** The Second Law of Gases, known as *Charles's Law,* defines the relations between pressure, volume, and temperature of a gas. This law states that, *at a constant pressure, the volume of a gas is proportional to its absolute temperature*; or, *at a constant volume, the pressure of a gas is proportional to its absolute temperature.* In other words, the product of pressure and volume in a gas is proportional to its absolute temperature.

Let $P$ = Pounds pressure per square inch, absolute;
$V$ = Volume in cubic feet;
$T$ = Absolute temperature.

Then,

$$PV = KT; \text{ or, } PV \div T = K.$$

The value of the constant $K$, for air, is found as follows: In the discussion of Boyle's law, it was shown that $PV = C = 182.08$ for 32° F. In this case the absolute temperature is $32 + 461 = 493°$ absolute. Substituting these figures in the above equation, it becomes $182.08 \div 493 = K = .3693$.

**Phenomena of Compression.** *Thermodynamics* is the science of the relation between heat and energy. It is based on two fundamental laws, only the first of which has a bearing on the discussion of compressed air.

The First Law of Thermodynamics states that *where heat is converted into mechanical energy, or mechanical energy is converted into heat, the quantity of heat is exactly equivalent to the amount of mechanical energy.*

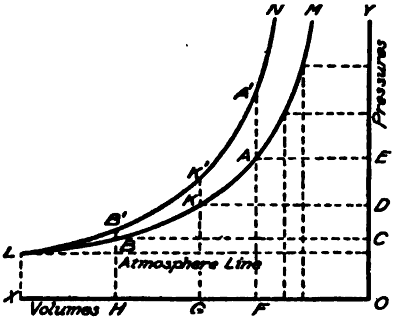


Fig. 2. Isothermal Curve and Adiabatic Curve Illustrating Charles's Law of Pressures, Volumes, and Temperatures.

This law demands that when work is done upon a volume of air in compressing it from a lower to a higher pressure, a quantity of heat must be developed exactly equivalent to the energy expended in compression.

In other words, when a volume of air is compressed to a higher pressure, all the work done upon that air volume is converted into heat; and that heat acts to increase the temperature of the air volume, whether the process of compression be slow or fast.

All modern air-compressors consist fundamentally of a cylinder with a moving piston; and the air is compressed by crowding it into a constantly diminishing space in the cylinder in advance of the moving piston. According to Boyle's law, as soon as the volume is diminished, the pressure must increase.

It has just been stated that the compression of air produces an increase in its temperature; and according to Charles's law, this increased temperature, acting on the volume of air in the compressing cylinder (where the volume at any instant is constant), produces an increase of pressure.

In the ordinary process of air-compression, therefore, two elements are at work toward the production of a higher pressure—*first,* the reduction of volume by the advancing piston; *second,* the increasing temperature due to the increasing pressure corresponding to the reduced volume.

**Adiabatic Compression.** This is compression without the removal or escape of any of the heat produced by the compression process.

It has just been shown that the pressure of an air volume during compression in a cylinder is at any moment the sum of two different pressures. The first pressure is that due to the reduction of volume; and since temperature does not enter into this increment of pressure, the line showing the course of this increasing pressure will be an isothermal curve. The second pressure is due to the increasing temperature, and is to be added, at any given moment or position of the piston, to the first pressure.

In Fig. 2, the curve $A\ K\ B$ is a curve of isothermal compression. The curve $A'\ K'\ B'$ is a curve of adiabatic compression, and is found by adding to the lines $F\,A$, $G\,K$, and $H\,B$, representing pressures due to decreased volume, the lines $AA'$, $KK'$, and $BB'$, representing pressures due to the increase in temperature corresponding to the pressures $FA$, $GK$, and $HB$.

The formula for this curve is:

$$P_1V_1^N = P_2V_2^N = P_3V_3^N\text{, etc.,}$$

the exponent N providing for the heat of compression. The method by which the value of N is found has no place in this paper. It varies for different gases, and for air it is usually taken at 1.41. The equation of the adiabatic curve for air, therefore, is:

$$P_1V_1^{1.41} = P_2V_2^{1.41} = P_3V_3^{1.41}\text{, etc.}$$

$P_1$, $P_2$, $P_3$, etc., represent absolute pressures; $V_1$, $V_2$, $V_3$, etc., represent the corresponding volumes in adiabatic compression. This is simply another form of expressing Charles's law in its relation to air.

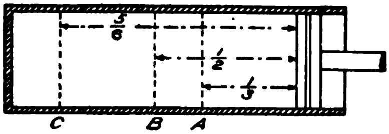


Fig. 3. Diagram Showing Effect of Heat of Compression

**Pressures and Volumes in Compression.** Assume a compressing cylinder (Fig. 3) with a cross-section of 1 square foot, and a piston travel or stroke of 3 feet. Let this be filled with air at atmospheric pressure of 14.7 lbs. absolute, and 60° F. temperature. If the piston be advanced to the position *A*, covering one-third of its stroke, the volume of air will be reduced one-third, and there will be three volumes crowded into the space of two, or the volume will be decreased in the ratio of 2 to 3; and since pressure varies inversely as the volume, according to Boyle's law, the pressure in the cylinder *due to the reduced volume* will be the original pressure increased in the ratio of 3 to 2, or $\frac{3}{2} \times 14.7 = 22.05$ lbs. absolute = 1½ atmospheres = 7.35 lbs. gauge.

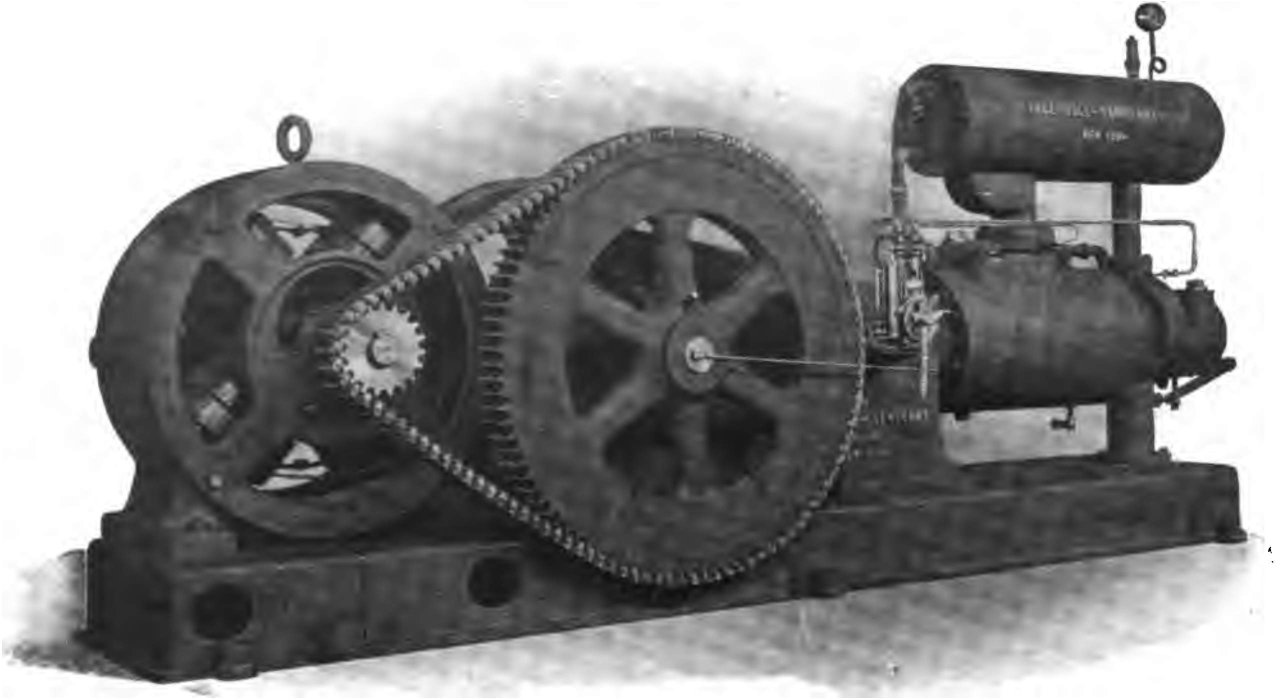
But a pressure gauge on the cylinder would at this time actually show a higher pressure than 7.35 lbs.; and the difference between this high pressure and the theoretically correct pressure of 7.35 lbs. is *due to the increase in temperature* produced in crowding three air volumes into the space of two.

Similarly, when the piston has advanced to *B*, or one-half stroke, the pressure should be 29.4 lbs. absolute, or 14.7 lbs. gauge; and at *C*, or $\frac{5}{6}$ stroke, the pressure should be 88.2 lbs. absolute, or 6 atmospheres, or 73.5 lbs. gauge. But it is actually found to be more than these figures in all cases, owing to the added pressure produced by the increased temperature.

If the cylinder were made of a non-heat-conducting material so that not a particle of the heat of compression could escape, the relations of pressures, volumes, and temperatures at any point of the stroke would be expressed by the equation for Charles's law, $PV = KT$. On the other hand, if it were possible to withdraw the heat of compression as fast as it was produced, the indicated gauge pressures would have been exactly as they theoretically should be, for compression would then follow the expression of Boyle's law, $PV = C$, the pressures varying exactly in inverse ratio with the volumes.

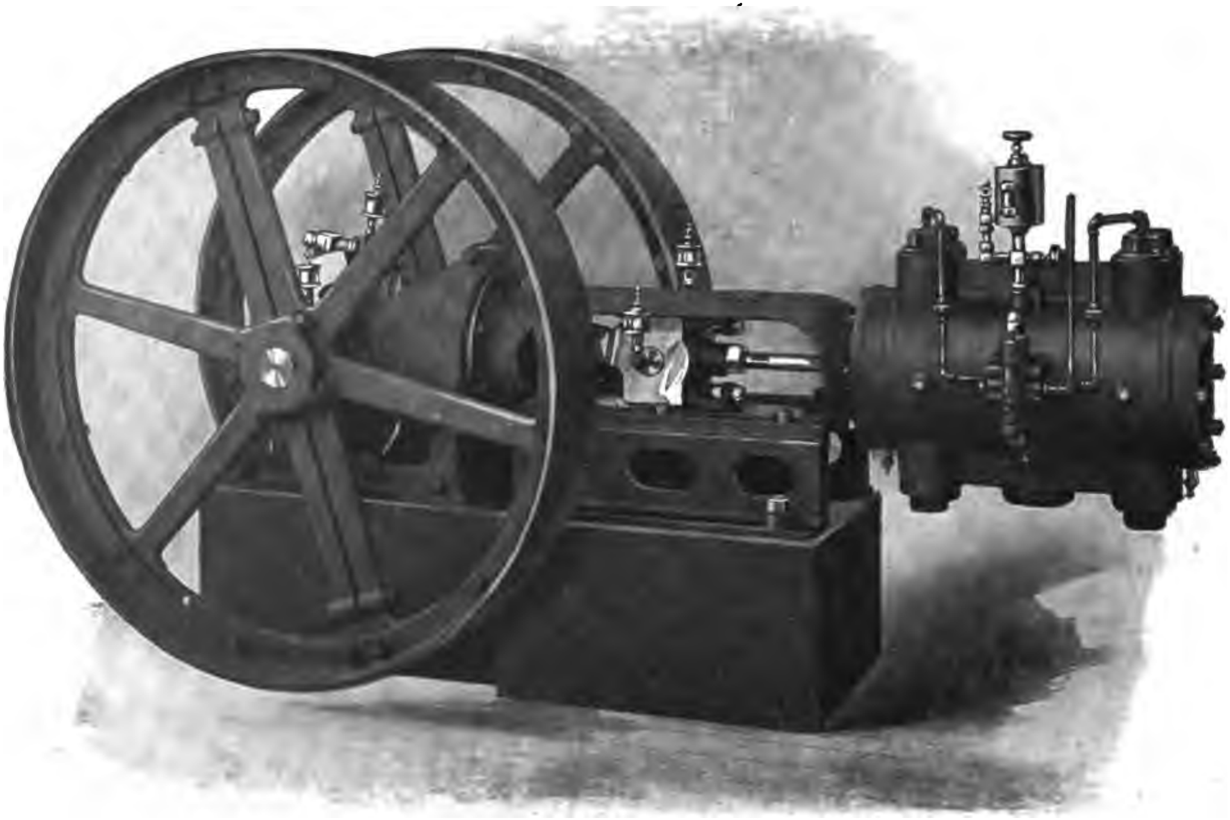
As a matter of fact, neither of these conditions is attained in practice. There is bound to be some escape of heat by conduction and radiation from the materials comprising the cylinder, piston, etc.; but it is not possible to remove all of this heat in practice. In actual air-compression work, means are adopted for removing the heat of compression as far as possible—which will be discussed later.

**Necessity for Cooling during Compression.** In the case just cited, the terminal pressure at $\frac{5}{6}$-stroke was found to be more than 73.5 lbs. gauge. If the piston is allowed to remain at this point until



**MOTOR-DRIVEN AIR COMPRESSOR.**

Morse Chain Company.



**BELT-DRIVEN AIR COMPRESSOR.**
**Sectionalized for Mule-back Transportation.**
**Sullivan Machinery Company.**

the temperature of the compressed air volume has dropped to 60° F., which was the temperature it had at the beginning of compression, and if it is assumed that there is no leakage, the final pressure will be found to be 73.5 lbs. gauge. If the discharge valves are then opened and the compressed air is discharged to a pipe-line, it will have a useful and stable pressure of 73.5 lbs. gauge, if the temperature of its surroundings is 60° F.

But to secure this useful and stable working pressure of 73.5 lbs., enough power had to be applied to produce the higher terminal pressure before cooling. In other words, there was a waste of power equivalent to that required to produce the difference in pressure between 73.5 lbs. and the higher but unstable terminal pressure. This waste of power, therefore, was due entirely to the heat generated in the air volume during compression.

Economical air-compression fundamentally demands, therefore, the removal of the heat of compression as fast as it is produced, so far as this is practically possible.

*Ideal air-compression* is compression along isothermal lines, following Boyle's law of pressures and volumes. The most uneconomical air-compression is compression along adiabatic lines, following Charles's law of pressures, volumes, and temperatures.

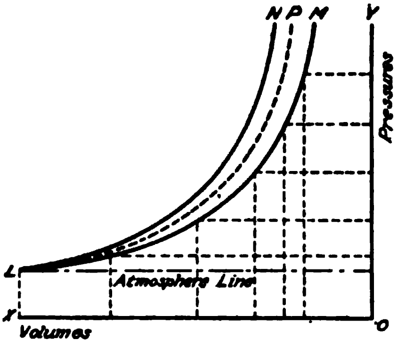


Fig. 4. Curve of Practical Compression in Relation to Isothermal and Adiabatic Curves.

It is possible in practice to avoid the latter extreme, but it is entirely impossible to realize the ideal expressed in the former extreme. The best air-compressor practice of to-day approaches that shown by the curve *LP* in Fig. 4, which is the mean between the adiabatic curve *LN* and the isothermal curve *LM*. Compressors of the highest refinement of design and of the highest economy, may show a curve following this intermediate line; but the simpler types of compressors, particularly in small sizes, will give a compression curve running closer to the adiabatic.

The question of compression economy is thus seen to be primarily one of adequate cooling during compression; and the accepted methods in use to-day will be discussed later in the proper place.

## TABLE II

### Final Temperatures Due to Adiabatic Compression to Various Gauge Pressures, from Initial Pressure of One Atmosphere, and from Various Initial Temperatures

(Richards)

| Final Pressure, Lbs. Gauge | Initial Temperature 0°F. | Initial Temperature 32°F. | Initial Temperature 60°F. | Initial Temperature 100°F. |
|---|---|---|---|---|
| 1 | 8 | 41 | 70 | 111 |
| 2 | 16 | 50 | 79 | 121 |
| 3 | 25 | 59 | 88 | 132 |
| 4 | 33 | 67 | 97 | 140 |
| 5 | 41 | 75 | 106 | 150 |
| 10 | 74 | 113 | 144 | 191 |
| 15 | 104 | 144 | 177 | 226 |
| 20 | 130 | 171 | 207 | 258 |
| 25 | 153 | 196 | 233 | 287 |
| 30 | 175 | 219 | 258 | 313 |
| 35 | 195 | 240 | 280 | 337 |
| 40 | 213 | 260 | 301 | 360 |
| 45 | 231 | 279 | 321 | 381 |
| 50 | 247 | 296 | 339 | 401 |
| 55 | 262 | 316 | 357 | 420 |
| 60 | 277 | 328 | 373 | 437 |
| 65 | 291 | 343 | 389 | 454 |
| 70 | 304 | 358 | 404 | 471 |
| 75 | 317 | 371 | 419 | 486 |
| 80 | 330 | 384 | 433 | 501 |
| 85 | 342 | 397 | 446 | 516 |
| 90 | 353 | 410 | 459 | 530 |
| 95 | 364 | 422 | 472 | 543 |
| 100 | 375 | 435 | 484 | 556 |
| 125 | 425 | 486 | 540 | 617 |
| 150 | 468 | 532 | 588 | 669 |
| 175 | 507 | 574 | 633 | 717 |
| 200 | 542 | 612 | 672 | 781 |

**Heating Effect of Compression.** While repeated reference has been made to the heat produced by compression, no definite values have as yet been assigned in these pages. Table II is accordingly presented, being copied from the book "Compressed Air," by courtesy of the author, Mr. Frank Richards. It shows the temperature corresponding to the various pressures in adiabatic compression of free air, from initial temperatures of 0°, 32°, 60°, and 100° F. Of course it need never be expected to find in practice these terminal temperatures after compressing to the pressures given. This table assumes that none of the heat of compression escapes. As a matter of fact, however, there is sure to be some loss of heat by conduction and radiation from the materials of which the compressor is built.

## TABLE III

### Loss of Work Due to Heat in Compressing Air from Atmospheric Pressure to Various Gauge Pressures by Simple and Compound Compression

(E. F. Schaefer)

Air in each cylinder at initial temperature of 60° F.

| Gauge Pressure | One-Stage | | Two-Stage | | Three-Stage | | Four-Stage | |
|---|---|---|---|---|---|---|---|---|
| | Percentage of Work Lost, in Terms of | | | | | | | |
| | Isothermal Compression | Adiabatic Compression | Isothermal Compression | Adiabatic Compression | Isothermal Compression | Adiabatic Compression | Isothermal Compression | Adiabatic Compression |
| 60 | 29.9 | 23.0 | 13.4 | 11.8 | 8.6 | 7.9 | 4.7 | 4.5 |
| 70 | 30.6 | 23.4 | 14.1 | 12.4 | 8.7 | 8.0 | 6.1 | 5.7 |
| 80 | 32.7 | 24.6 | 14.7 | 12.8 | 9.7 | 8.9 | 6.4 | 6.0 |
| 90 | 34.7 | 25.8 | 16.1 | 13.8 | 10.5 | 9.5 | 7.3 | 6.8 |
| 100 | 36.7 | 26.8 | 16.9 | 14.5 | 10.9 | 9.8 | 7.8 | 7.3 |
| 125 | 41.1 | 29.2 | 18.5 | 15.6 | 11.6 | 10.4 | 8.8 | 8.1 |
| 150 | 44.8 | 30.9 | 20.1 | 16.7 | 12.3 | 10.9 | 9.1 | 8.4 |
| 200 | 51.2 | 33.9 | 22.2 | 18.1 | 14.0 | 12.3 | 10.5 | 9.5 |
| 300 | 61.2 | 37.9 | 25.7 | 20.5 | 16.6 | 14.2 | 12.0 | 10.7 |
| 400 | 68.7 | 40.7 | 28.9 | 22.4 | 18.2 | 15.4 | 13.1 | 11.5 |
| 500 | 70.6 | 41.4 | 31.2 | 23.8 | 19.3 | 16.2 | 14.1 | 12.3 |
| 600 | 80.4 | 44.5 | 32.8 | 24.7 | 20.4 | 16.9 | 14.9 | 13.0 |
| 700 | 85.0 | 46.0 | 34.6 | 25.7 | 21.3 | 17.6 | 16.1 | 13.8 |
| 800 | 89.5 | 47.2 | 35.7 | 26.3 | 22.0 | 18.1 | 16.2 | 13.9 |
| 900 | 93.0 | 48.2 | 37.1 | 27.0 | 22.6 | 18.5 | 16.6 | 14.4 |
| 1,000 | 96.1 | 49.0 | 37.9 | 27.5 | 23.2 | 18.8 | 16.9 | 14.5 |
| 1,200 | | 50.7 | 40.3 | 28.8 | 24.8 | 19.9 | 17.7 | 15.0 |
| 1,400 | | 52.0 | 41.5 | 29.3 | 25.9 | 20.5 | 18.6 | 15.7 |
| 1,600 | | 53.1 | 43.5 | 30.3 | 26.5 | 20.9 | 19.2 | 16.1 |
| 1,800 | | 54.0 | 44.8 | 31.0 | 27.3 | 21.2 | 19.6 | 16.4 |
| 2,000 | | 55.0 | 45.8 | 31.4 | 27.5 | 21.5 | 19.9 | 16.5 |

**Loss of Work Due to Heat of Compression.** Table III, computed by E. F. Schaefer, and used here by courtesy of the Ingersoll-Rand Company, shows the magnitude of the losses in air-compression due to the heat of compression, the figures at all pressures being given for both single-, two-, three-, and four-stage compression. The values assume an initial temperature in all cylinders of 60° F.; and since this is a condition which would probably never be fully realized in practice, the actual losses may be assumed to be even larger than those here given.

**Compressed-Air Indicator Card.** The general theory of the indicator card in its relation to the steam engine may here be assumed to be known by the reader; but the importance of the subject of the compressed-air indicator card justifies a somewhat extended special treatment.

To avoid confusion on the part of those familiar only with the steam engine indicator card, it is to be remembered that in the air-compressor indicator card all the operations are exactly reversed. because the former is a record of expansions and the latter a record of compressions.

To be exact, the admission line of the steam card corresponds to the delivery or discharge line of the air card; the expansion line of the former, to the compression line of the latter; the exhaust or back-pressure line of the former, to the admission or intake line of the latter; and the compression line of the former, to the re-expansion line of the latter.

At the outset, let it be understood that an air-compressor card, to be a correct record of the machine's performance, must be taken only after the compressor has been run long enough to get thoroughly *warmed up*, so that all temperature conditions affecting compression have attained their normal running maximum—which is usually not until the machine has been running an hour or more.

An ideal indicator card from an air-compressor is set forth in Fig. 5. In this figure, *GH* is the line of zero pressure, drawn at the proper distance below the atmospheric line *EF*, according to the scale. *DA* is the admission or intake line; *AB*, the compression line; *BC*, the delivery or discharge line; and *CD*, the re-expansion line. The admission line *DA* is here drawn at an exaggerated distance below the atmospheric line *EF*, simply for the sake of clearness. In the best practice, these two lines so nearly coincide as to be hardly distinguishable. The line *CO* having been dropped vertically from the point *C*, the distance *EO* represents the stroke of the piston. The line *CO* (or *FR*) represents the maximum gauge pressure.

The rectangle *E N C O* represents the actual volume displaced by the piston in moving from *E* to *O*. This, however, is not the total capacity of the cylinder, for, at the end of the stroke, there remains a small volume, represented by the rectangle *O C K J*, in the clearance space of the cylinder. This small volume is, of course, compressed to maximum pressure. On the return stroke, it expands to atmosphere along the curve *CD*, its expansion occurring so quickly as to be practically adiabatic. Thus the total volume of air acted upon by the piston is represented by the rectangle $E\ N\ K\ J$.

The distance $OJ$ (or $CK$), representing the clearance percentage, is found as follows: Assume that the scale of the spring used in taking the card is 30, and the terminal pressure is 70 lbs. gauge. The volume of air in the clearance space, after expansion, may be found from column 5 of Table VI (page 21). Here the volume of air expanding without cooling, or adiabatically, from 70 lbs. to atmosphere, is found to be 0.288, as compared with the unit-volume. Let $X$ = the distance $OJ$. Then,

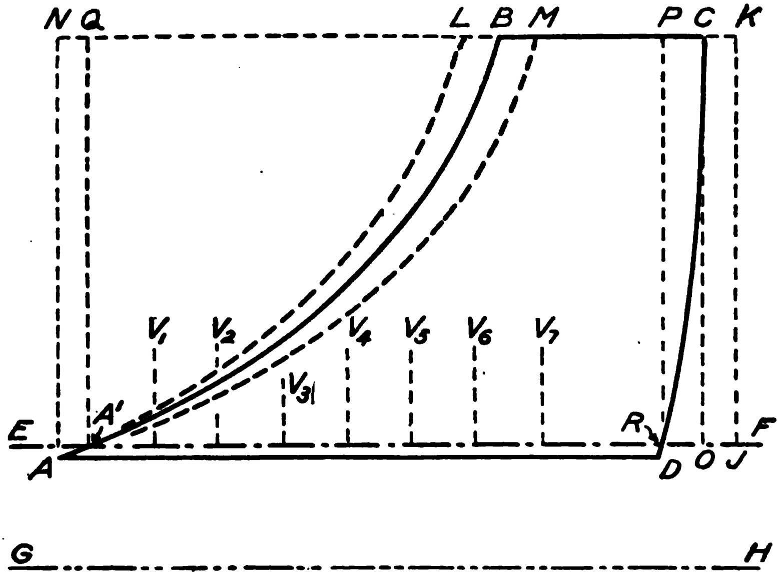


Fig. 5. An Ideal Air-Compressor Indicator Card.

$$X : DO + X = 0.288 : 1.00.$$

The length $DO$ can be measured from the diagram, and may be here assumed as 0.25 inch. Then,

$$X : 0.25 + X = 0.288 : 1.00;$$

whence,

$$X = 0.101 \text{ inch (say 0.1 inch).}$$

It will be noted that, since compression began a little below atmosphere, the compression line $AB$ intersects the atmosphere line $EF$ at $A'$, which, being the point at which compression from atmosphere begins, may be taken as the initial point from which the entire cycle of operations represented in the diagram starts.

Erecting the perpendicular *A′Q*, the rectangle *A′ Q K J* represents the total volume of air *at atmospheric pressure* upon which power is exerted for compression.

With the piston at the end of its stroke, there still remains the air at terminal or discharge pressure in the clearance space represented by *O C K J*. On the return stroke, this air expands to atmosphere, the re-expansion line *CD* intersecting the atmosphere line at *R*. This volume, while not delivered or discharged, and having required an expenditure of power to compress it, gives back this power by its re-expansion, helping the piston on its return stroke.

*Clearance in an air-compressor cylinder thus represents, not a loss of power, but a loss of capacity.*

With this understanding of the quantities represented on the indicator card, an investigation of its possibilities as a source of information on the performance of a machine may be undertaken.

In computing the work done in the cylinder, and the mean effective pressure (M.E.P.) throughout the stroke, the entire area enclosed by the diagram *A B C D* must be used. This area is to be measured by the means usually employed in such cases—that is, by a planimeter or by the use of ordinates. This area, of course, represents only the work done in one stroke, representing only one end of a double-acting compression cylinder. It is much more accurate to have deductions based upon the mean of cards from both ends of the cylinder; but the present card will serve as a guide for all cases.

The area *AA′ RD,* lying below the atmosphere line, represents the resistance on the return stroke; but it may be assumed that the cards from both ends of the cylinder are similar, and this area may therefore be safely considered in connection with the area above the atmosphere line for the single stroke.

*Computing the I. H. P. of the Air Cylinder.* The *indicated horse-power* (I. H. P.) of a compressor cylinder is figured in the same manner as for a steam cylinder. Assume that the M.E.P. has been found to be 30 lbs., and that the card under examination was taken from a compressor having a 24-inch cylinder and a 30-inch stroke, running at 75 revolutions per minute (R.P.M.). Then

$P = 30$ lbs.; $L = 2 \times 30 \div 12 = 5$ ft.; $A = 24^2 \times .7854 = 452.4$ sq. in.; $N = 75$ R.P.M. Therefore,

$$\text{I.H.P.} = P \times L \times A \times N \div 33{,}000 = 154.23 \text{ H.P}$$

This, it will be noted, is simply the horse-power representing the work done in compressing the air. It takes no account of friction or other losses.

*Computing the Mechanical Efficiency of a Compressor.* The *mechanical efficiency* of an air-compressor is the ratio of the power developed in the air cylinder to the power applied in the steam cylinder. In other words, it is the ratio of the I.H.P. of the air cylinder to the I.H.P. of the steam cylinder, if the compressor is steam-driven. The difference between these two represents the power required to keep the machine going, overcome friction, etc. In a power-driven compressor, the mechanical efficiency is the ratio of the I.H.P. of the air cylinder to the *brake horse-power* (B.H.P.) applied to the driving wheel or shaft.

A comparison of the steam and air cards taken simultaneously from a steam-driven compressor is thus seen to show exactly what percentage of the power applied is used in overcoming the friction of the machine, and what percentage is used in actually compressing the air. These proportions can also be determined exactly in the case of a power-driven compressor where the applied power can be actually measured, as by a brake or other dynamometer. There is probably no other machine in everyday use in which the mechanical efficiency can be determined so accurately as in an air-compressor.

*Determining the Volumetric Efficiency of a Compressor.* The *volumetric efficiency* of an air-compressor is the ratio of the actual volume of air compressed and discharged, to the volume theoretically displaced by the piston in a given time. This is shown with great accuracy by the air-compressor indicator card.

The point $A'$ (Fig. 5), where the compression line crosses the atmosphere line, indicates that the cylinder is full of air at atmospheric pressure at this point in the stroke. The distance $EA'$ represents the portion of the stroke required to compress to atmospheric pressure the rarified air indicated by the intake line. This distance is therefore to be deducted from the full stroke $EO$ as non-effective in the compression and delivery of atmospheric air under pressure. This discussion emphasizes the importance of bringing the intake line as near as possible to the atmosphere line, as the distance *EA′* evidently increases as the intake line drops below atmosphere.

At the end of the stroke, the clearance air remains at terminal pressure, and on the return stroke re-expands to atmosphere, its re-expansion line crossing the atmospheric line at the point *R*. Evidently atmospheric air cannot enter through the inlet valves of a compressor until this air in the clearance space has expanded to atmospheric pressure. The distance *RO*, therefore, represents the proportion of the stroke occupied in compressing and re-expanding the clearance air. It is non-effective in the admission of atmospheric air, and it also must be deducted from the full stroke *EO*.

In the diagram (Fig. 5), *EO* measures approximately 4 inches; *RO* is about $\frac{1}{4}$ inch, and *EA′* about $\frac{3}{16}$ inch. The ratio of the effective stroke to actual stroke is thus seen to be $3\frac{9}{16}:4$, or 0.89. This ratio is evidently also the ratio of volumes, and therefore the volumetric efficiency represented by this card is 89 per cent. This is lower than is found in the best practice; but in this assumed card many distances are exaggerated for the sake of greater clearness.

It would at first seem that this method would give exactly the volumetric efficiency of the cylinder under examination; but one serious shortcoming of the indicator card as a record of performance must here be noted. While the method just described gives accurately the ratio of the air volume compressed and discharged to the cylinder displacement, it does not take into consideration the temperature of the air. It is not usually safe to assume that the air admitted to the cylinder is actually of the density of the surrounding free air, for it may be heated, and expanded accordingly, in entering the cylinder through heated ports and valves. An example will illustrate this. If the atmospheric temperature is 70° F., and the air in entering the cylinder is heated to 120 ° F., the ratio of densities in the two cases, from Table I (page 4), will be 0.0685:0.075, or 0.913. This means that only 0.913 of the air volume apparently compressed is actually compressed. More than this, it means that the full power as shown by the steam card is being used to compress this reduced volume of air.

There is no way of determining exactly the initial temperature *in the cylinder*; and the best that can be done, the conditions being understood, is to arrange as far as possible to avoid the heating of the intake air. This matter will receive more attention in a later section where cooling is discussed.

Much can be done, however, by seeing that the air is delivered *to the compressor intake* as cool as possible. The air of the average engine room is heated, and should not be admitted to the cylinder. It is well worth while to lay an inlet duct from the compressor intake to where the coolest possible air can be had—for example, outside the engine room, from the north side of a building, etc. Table IV shows the importance of this point. An examination of its figures reveals the fact that for every 10 degrees reduction in intake temperature, there is a gain of practically 2 per cent in capacity and volumetric efficiency. This is a small economy often overlooked, though its only cost is that of laying a suitable intake duct or tube to the right place, making it, of course, large enough so that the air flows freely and no *suction load* is imposed upon the piston.

### TABLE IV

**Effect of Various Initial or Intake Temperatures of Air on the Capacity of the Compressor and the Efficiency of Compression**

| Initial Temperature | | Relative Capacity and Efficiency | Initial Temperature | | Relative Capacity and Efficiency |
|---|---|---|---|---|---|
| Fahrenheit | Absolute | | Fahrenheit | Absolute | |
| −20 | 441 | 1.180 | 70 | 531 | .980 |
| −10 | 451 | 1.155 | 80 | 541 | .961 |
| 0 | 461 | 1.130 | 90 | 551 | .944 |
| 10 | 471 | 1.104 | 100 | 561 | .928 |
| 20 | 481 | 1.083 | 110 | 571 | .912 |
| 30 | 491 | 1.061 | 120 | 581 | .896 |
| 32 | 493 | 1.058 | 130 | 591 | .880 |
| 40 | 501 | 1.040 | 140 | 601 | .866 |
| 50 | 511 | 1.020 | 150 | 611 | .852 |
| 60 | 521 | 1.000 | 160 | 621 | .838 |

*Effect of Clearance Space.* There must of necessity be some clearance between the piston and cylinder heads, merely on the grounds of safety. In addition, there are usually spaces left after the inlet and discharge valves are closed—large or small according to the mechanical design. These spaces which, in the aggregate, constitute the clearance space, are filled with air at terminal pressure at the end of the stroke; and when the return stroke begins, this air re-expands to atmosphere or intake pressure. This air is compressed and re-expanded at each end of the cylinder, once each revolution or double stroke. Power is spent to compress it; and on expansion, it gives back practically all this power as work applied in helping the piston on the return stroke. In this respect, therefore, clearance does not represent a loss of power.

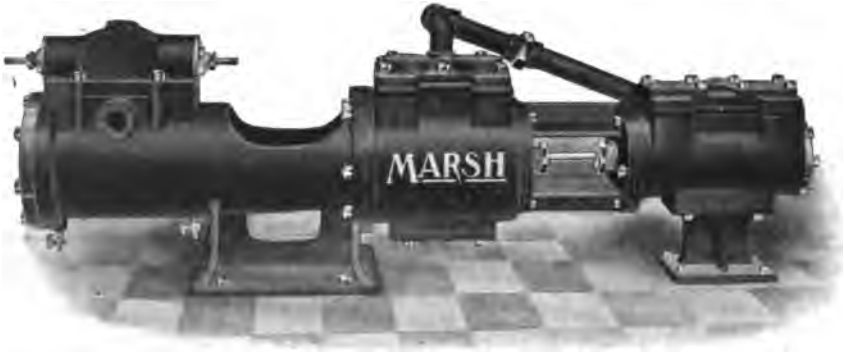
But, as has been stated, no atmospheric air can enter the cylinder until the clearance air is expanded to atmosphere. Thus a portion of the admission stroke is really no admission at all, and a full cylinder of air is never admitted in practice. Clearance, therefore, does mean a loss of capacity, and should be kept down to the lowest possible percentage. Furthermore, since air-compressors are rated by piston displacement, and since the efficiencies are based on this rating rather than on the actual delivered volume (which can seldom be measured), the loss of capacity due to clearance is really in effect a proportionate loss of power, if the theoretical power is figured according to piston displacement.

*Determining the Compression Efficiency.* The *compression efficiency* of an air-compressor is the ratio of the minimum power theoretically required to produce a given pressure to the power actually expended in the air cylinder in producing that pressure. Since the minimum power is that of isothermal compression, the compression efficiency is the ratio of the isothermal horse-power to the indicated horse-power as registered by the indicator card.

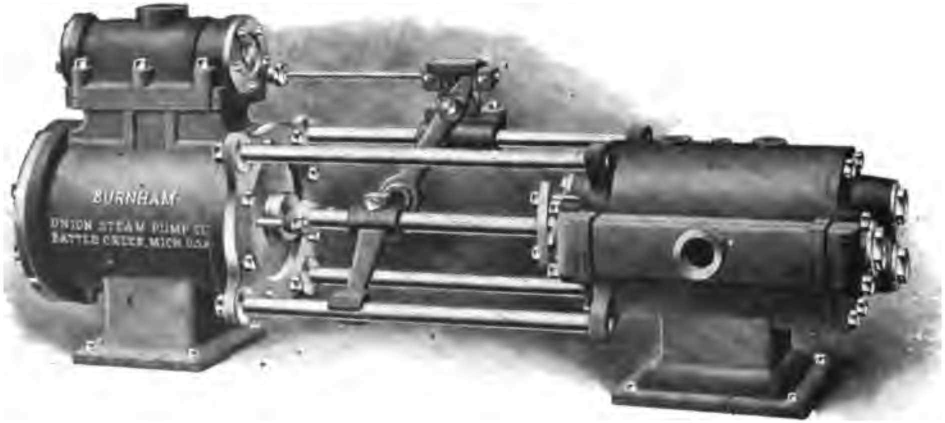
The power developed or expended in the air cylinder is proportional to the M.E.P., since, in the horse-power formula $HP = PLAN \div 33{,}000$, all factors except $P$ may be considered as constant throughout the stroke. Accordingly the ratio of powers in computing compression efficiency is the same as the ratio of M.E.P.'s for isothermal conditions and for the actual conditions as the indicator card reveals them.

The simplest way, therefore, to figure the compression efficiency of a compressor, is to compare the M.E.P. of its indicator card with the M.E.P. corresponding to the given pressure, as given in column 6 of Table VI (page 21). It is to be borne in mind, however, that Table VI is correct only for sea-level conditions.

*Laying Out the Isothermal and Adiabatic Curves.* The method just described for determining the compression efficiency gives all



**MARSH TWO-STAGE AIR COMPRESSOR**
American Steam Pump Company, Battle Creek, Mich.



BURNHAM AIR COMPRESSOR
Union Steam Pump Co., Battle Creek, Mich.

the information ordinarily required; but it is sometimes useful, as a graphic demonstration, to lay out the isothermal and adiabatic lines on the indicator card in their proper relations to the lines of the diagram. The method is as follows:

In isothermal compression, $PV = C$, a constant; or $P_1 V_1 = P_2V_2 = P_3V_3$, etc. Transposing, $P_2 = P_1 (V_1 \div V_2)$; $P_3 = P_1 (V_1 \div V_3)$, etc. Referring once more to Fig. 5, it will be remembered that the rectangle *A'QKJ* represents the total volume of atmospheric air undergoing compression, and the division of this rectangle by vertical lines would indicate proportionate volumes at corresponding points of stroke. Let these lines $V_1$, $V_2$, $V_3$, etc., be drawn, dividing this rectangle into ten equal parts. Let the volume represented by *A'J* be 1; then the volume represented by $V_1J$ will be 0.9; by $V_2J$, 0.8; etc. If the initial pressure of the card is 14.7 lbs. absolute, this will be $P_1$. The values in the above equation, then, become:

$P_2 = 14.7\ (1.0/0.9) = 16.317$ lbs. absolute.
$P_3 = 14.7\ (1.0/0.8) = 18.375$ lbs. "
$P_4 = 14.7\ (1.0/0.7) = 20.874$ lbs. "

In like manner the absolute pressures at any percentage of the stroke may be figured; and if these pressures on the proper scale are laid off above the atmospheric line on the corresponding vertical lines of the diagram, a curve joining the points thus determined will be the *isothermal curve* for that diagram.

For any other initial pressure than 14.7 lbs., the method is exactly the same, with the exception that this new value of $P_1$ is to be substituted in the formula. Further precaution must be taken to see that the atmospheric line is properly located in relation to the admission line.

To plot the adiabatic curve, the method is the same as for the isothermal, except that the mathematics involved demands the use of logarithms because of the fractional exponent. The formula to be used in this case is $P_2 = P_1 (V_1 / V_2)^{1.41}$; $P_3 = P_1 (V_1 / V_3)^{1.41}$; etc. This equation is solved by logarithms thus:

$$\log. P_2 = \log. P_1 + \left\{ (\log. V_1 - \log. V_2) \times 1.41 \right\}$$

Pressures thus obtained for the various percentages of stroke are laid off on the corresponding vertical lines, and the points thus determined are joined by a line, which is the *adiabatic curve.*

## TABLE V

### Multipliers for Determining the Isothermal and Adiabatic Pressures in Air-Compression from any Initial Pressure, at Various Percentages of Stroke

| Percentage of Stroke | Isothermal | Adiabatic | Percentage of Stroke | Isothermal | Adiabatic |
|---|---|---|---|---|---|
| 0 | 1.000 | 1.000 | 50 | 2.000 | 2.657 |
| 5 | 1.052 | 1.074 | 55 | 2.222 | 3.083 |
| 10 | 1.111 | 1.160 | 60 | 2.500 | 3.640 |
| 15 | 1.176 | 1.256 | 65 | 2.857 | 4.394 |
| 20 | 1.250 | 1.369 | 70 | 3.333 | 5.460 |
| 25 | 1.333 | 1.499 | 75 | 4.000 | 7.052 |
| 30 | 1.428 | 1.652 | 80 | 5.000 | 9.672 |
| 35 | 1.538 | 1.835 | 85 | 6.666 | 14.501 |
| 40 | 1.666 | 2.054 | 90 | 10.000 | 25.704 |
| 45 | 1.818 | 2.323 | 95 | 20.000 | 68.305 |

Table V gives the multipliers for determining the isothermal and the adiabatic pressures at the percentages of the stroke given, *per pound of initial pressure.* These multipliers are, for isothermal pressures, the value of $V_1 / V_2$; and for adiabatic pressures, the value of $(V_1 / V_2)^{1.41}$, etc. To use this table in plotting curves, simply multiply the atmospheric or intake pressure in pounds absolute by these constants; lay out the pressures thus secured at the proper percentages of stroke; and join the points by a curve.

**Relation of Pressures, Volume, and Temperatures.** Table VI (page 21) is reproduced by courtesy of Mr. Frank Richards, from his work on "Compressed Air," and was prepared by him for use in working up indicator cards and in other compressed-air computations. The basis of these figures is air at atmospheric pressure of 14.7 lbs. and a temperature of 60° F.; and the values in the table start with the air in this condition.

Column 1 lists the *gauge pressure,* which would be the ordinary working pressure of the air after compression.

Column 2 gives the corresponding *absolute pressure,* and the figures are found by adding 14.7 lbs. to the gauge pressure in column 1.

Column 3 gives the *pressure in atmospheres,* and is found by dividing the corresponding figure in column 2 by 14.7.

## TABLE VI

### Volumes, Mean Pressures, Temperatures, etc., in Compressing Air from One Atmosphere and 60° F.

(Richards)

| 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Gauge Pressure | Absolute Pressure | Pressure in Atmospheres | Volume with Air at Constant Temperature | Volume with Air not Cooled | M.E.P. per Stroke, Air at Constant Temperature | M.E.P. per Stroke, Air not Cooled | M.E.P. per Stroke, Average Conditions of Practice | M.E.P. for Compression Only, Air at Constant Temperature | M.E.P. for Compression Only, Air not Cooled | M.E.P. for Compression Only, Average Conditions of Practice | Final or Terminal Temperature, Air not Cooled |
| 0 | 14.7 | 1.000 | 1.0000 | 1.00 | 0.00 | 0.000 | 0.000 | 0.00 | 0.00 | 0.000 | 60 |
| 1 | 15.7 | 1.068 | .9363 | .95 | .96 | .975 | .967 | .43 | .44 | .435 | 71 |
| 2 | 16.7 | 1.136 | .8803 | .91 | 1.87 | 1.91 | 1.89 | .95 | .96 | .955 | 80.4 |
| 3 | 17.7 | 1.204 | .8305 | .876 | 2.72 | 2.8 | 2.76 | 1.4 | 1.41 | 1.405 | 88.9 |
| 4 | 18.7 | 1.272 | .7861 | .84 | 3.53 | 3.67 | 3.60 | 1.84 | 1.86 | 1.85 | 98 |
| 5 | 19.7 | 1.34 | .7462 | .81 | 4.3 | 4.5 | 4.40 | 2.22 | 2.26 | 2.24 | 106 |
| 10 | 24.7 | 1.68 | .5952 | .69 | 7.62 | 8.27 | 7.94 | 4.14 | 4.26 | 4.20 | 145 |
| 15 | 29.7 | 2.02 | .495 | .606 | 10.33 | 11.51 | 10.92 | 5.77 | 5.99 | 5.88 | 178 |
| 20 | 34.7 | 2.36 | .4237 | .543 | 12.62 | 14.4 | 13.52 | 7.2 | 7.58 | 7.30 | 207 |
| 25 | 39.7 | 2.7 | .3703 | .494 | 14.59 | 17.01 | 15.80 | 8.49 | 9.05 | 8.77 | 234 |
| 30 | 44.7 | 3.04 | .3289 | .4638 | 16.34 | 19.4 | 17.87 | 9.66 | 10.39 | 10.02 | 255 |
| 35 | 49.7 | 3.381 | .2957 | .42 | 17.92 | 21.6 | 19.76 | 10.72 | 11.59 | 11.15 | 281 |
| 40 | 54.7 | 3.721 | .2687 | .393 | 19.32 | 23.66 | 21.49 | 11.7 | 12.8 | 12.25 | 302 |
| 45 | 59.7 | 4.061 | .2462 | .37 | 20.52 | 25.59 | 23.05 | 12.62 | 13.95 | 13.28 | 321 |
| 50 | 64.7 | 4.401 | .2272 | .35 | 21.79 | 27.39 | 24.59 | 13.48 | 15.05 | 14.26 | 339 |
| 55 | 69.7 | 4.741 | .2109 | .331 | 22.77 | 29.11 | 25.94 | 14.3 | 15.98 | 15.14 | 357 |
| 60 | 74.7 | 5.081 | .1968 | .3144 | 23.84 | 30.75 | 27.29 | 15.05 | 16.89 | 15.97 | 375 |
| 65 | 79.7 | 5.423 | .1844 | .301 | 24.77 | 31.69 | 28.23 | 15.76 | 17.88 | 16.82 | 389 |
| 70 | 84.7 | 5.762 | .1735 | .288 | 26.00 | 33.73 | 29.86 | 16.43 | 18.74 | 17.58 | 405 |
| 75 | 89.7 | 6.102 | .1639 | .276 | 26.65 | 35.23 | 30.94 | 17.09 | 19.54 | 18.31 | 420 |
| 80 | 94.7 | 6.442 | .1552 | .267 | 27.33 | 36.6 | 31.96 | 17.7 | 20.5 | 19.10 | 432 |
| 85 | 99.7 | 6.782 | .1474 | .2566 | 28.05 | 37.94 | 32.99 | 18.3 | 21.22 | 19.76 | 447 |
| 90 | 104.7 | 7.122 | .1404 | .248 | 28.78 | 39.18 | 33.98 | 18.87 | 22.0 | 20.43 | 459 |
| 95 | 109.7 | 7.462 | .134 | .24 | 29.53 | 40.4 | 34.96 | 19.4 | 22.77 | 21.08 | 472 |
| 100 | 114.7 | 7.802 | .1281 | .232 | 30.07 | 41.6 | 35.83 | 19.92 | 23.43 | 21.67 | 485 |
| 105 | 119.7 | 8.142 | .1228 | .2254 | 30.81 | 42.78 | 36.79 | 20.43 | 24.17 | 22.30 | 496 |
| 110 | 124.7 | 8.483 | .1178 | .2189 | 31.39 | 43.91 | 37.65 | 20.9 | 24.85 | 22.87 | 507 |
| 115 | 129.7 | 8.823 | .1133 | .2129 | 31.98 | 44.98 | 38.48 | 21.39 | 25.54 | 23.46 | 518 |
| 120 | 134.7 | 9.163 | .1091 | .2073 | 32.54 | 46.04 | 39.29 | 21.84 | 26.2 | 24.02 | 529 |
| 125 | 139.7 | 9.503 | .1052 | .202 | 33.07 | 47.06 | 40.06 | 22.26 | 26.81 | 24.53 | 540 |
| 130 | 144.7 | 9.843 | .1015 | .1969 | 33.57 | 48.1 | 40.83 | 22.69 | 27.42 | 25.05 | 550 |
| 135 | 149.7 | 10.183 | .0981 | .1922 | 34.05 | 49.1 | 41.57 | 23.08 | 28.05 | 25.56 | 560 |
| 140 | 154.7 | 10.523 | .095 | .1878 | 34.57 | 50.02 | 42.29 | 23.41 | 28.66 | 26.03 | 570 |
| 145 | 159.7 | 10.864 | .0921 | .1837 | 35.09 | 51.0 | 43.04 | 23.97 | 29.26 | 26.61 | 580 |
| 150 | 164.7 | 11.204 | .0892 | .1796 | 35.48 | 51.89 | 43.68 | 24.28 | 29.82 | 27.05 | 589 |
| 160 | 174.7 | 11.88 | .0841 | .1722 | 36.29 | 53.65 | 44.97 | 24.97 | 30.91 | 27.94 | 607 |
| 170 | 184.7 | 12.56 | .0796 | .1657 | 37.2 | 55.39 | 46.25 | 25.71 | 32.03 | 28.87 | 624 |
| 180 | 194.7 | 13.24 | .0755 | .1595 | 37.96 | 57.01 | 47.48 | 26.36 | 33.04 | 29.70 | 640 |
| 190 | 204.7 | 13.92 | .0718 | .154 | 38.68 | 58.57 | 48.62 | 27.02 | 34.06 | 30.54 | 657 |
| 200 | 214.7 | 14.6 | .0685 | .149 | 39.42 | 60.14 | 49.78 | 27.71 | 35.02 | 31.36 | 672 |

Column 4 gives the volumes (compared with initial volume of 1) after *isothermal compression* to the pressure given. Its values assume that the heat of compression has been removed as fast as produced, or that the air has been allowed to cool after compression to its initial temperature of 60° F.

Column 5 presents the volumes after *adiabatic compression* to the stated pressures, assuming that all the heat of compression is allowed to remain in the air.

Column 6 gives the M.E.P. on the air-piston during the stroke of compression, assuming that compression is *isothermal,* the air remaining at 60° F. throughout the stroke. The figures in this column represent the ideal of air-compression, and can never be attained in practice, since no cooling device is sufficiently effective. They are, however, the standard with which the results of actual practice are to be compared, in figuring compression efficiency.

Column 7 lists the M.E.P. on the air-piston during the stroke in *adiabatic* compression to the corresponding gauge pressures, and assumes that no cooling of the air whatever occurs during compression. Since there is always some loss of heat, however, these figures are not found in practice. If column 6 presents the ideal toward which practice aims, column 7 is equally important as the extreme to be avoided as far as possible.

Column 8 gives figures which are the mean or average of those in columns 6 and 7. In other words, these are the M.E.P.'s closely approximating those found in actual work with the best air-compressors dealing with these pressures. Small high-speed machines will probably not equal these values. An indicator card showing these M.E.P.'s, or very near to them, may be considered a very good card.

Column 9 gives the M.E.P. during the *compression* part of the stroke, assuming isothermal conditions. The actual process in an air cylinder may be divided into two parts—*first,* the compression of the air volume from atmosphere to terminal pressure; *second,* the delivery or discharge of this air at terminal pressure from the cylinder. Columns 6, 7, and 8 have to do with the entire process, and cover both compression and delivery. Columns 9, 10, and 11, on the contrary, have to do only with the first part of the process.

Column 10 lists the M.E.P. during the compression part of the stroke, assuming adiabatic compression.

Column 11 gives the M.E.P.'s which are the mean or average of those in columns 9 and 10, and which represent very fairly the ordinary results found in practice as derived from indicator cards of strictly high-grade compressors.

Column 12 gives the final or terminal temperatures after *adiabatic* compression to the pressures listed. The inevitable escape of heat during compression, even where no cooling devices are used, makes these figures as they appear here higher than they are ever found in actual work; but their magnitude emphasizes the importance of using some means of cooling to avoid destructive temperatures in air-compression.

**Indicator Cards from Multi-Stage Compressors.** The discussion thus far has dealt only with conditions involving compression to the required terminal pressure in a single cylinder; but, as will be seen later, the modern tendency in compressor design is toward compound or multi-stage compression, in which the compression process is divided between two, three, or even four cylinders. While in such cases the performance of each cylinder is exactly recorded on the indicator card from that cylinder, still it is sometimes desirable to show the *combined card*, presenting in one diagram the conditions of compression in all the air cylinders of the machine.

The general method is the same as in the case of steam-engine indicators, for making combined steam-engine diagrams, with which the reader is assumed to be familiar. The low-pressure card is left at its original size, and the cards of the other cylinders are reduced to the same scale by altering the length (representing volume) in proportion to the ratio of cylinder areas, and the height (representing pressures) in proportion to the scale of the springs used in taking the cards. The isothermal and adiabatic lines are then drawn, and the phenomena of compression in all cylinders are graphically presented in one diagram.

**Some Representative Indicator Cards.** Figs. 6 to 14, inclusive, are reproductions of indicator cards taken from the air ends of machines of various types and under various conditions. They are here introduced to give an idea of what may be expected from air-compressors in everyday work, as these cards are all taken from machines running under practical conditions.

Fig. 6 is a card from a single-stage straight-line compressor with a cylinder 26 inches in diameter and 30 inches stroke, running at a speed of 88 R.P.M. and compressing to 61 lbs. gauge. The atmospheric pressure is 14.7 lbs. absolute; and it will be noted that, while the intake line starts on the atmospheric line, it drops below toward the end of the admission stroke, the intake pressure at the end of the stroke being 13.5 lbs. absolute. This would indicate an insufficient inlet-valve area, or that a partial vacuum or rarification of the intake air was produced. The compression line runs rather closer than usual to the adiabatic. The waves in the discharge line are due largely to vibration in the springs of the poppet discharge valves, the variations being amplified by the vibration of the indicator spring. The re-expansion line is good. The volumetric efficiency shown by this card is 87 per cent. . The I. H. P. of the card is 201.5; and the efficiency, compared with perfect isothermal single-stage compression, is 93 per cent.

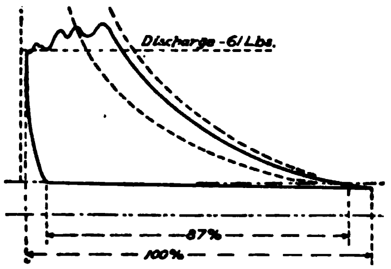


Fig. 6. Single-Stage Air-Compressor Indicator Card.

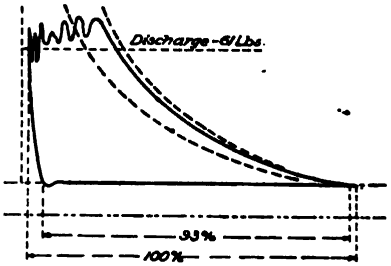


Fig. 7. Single-Stage Air-Compressor Indicator Card.

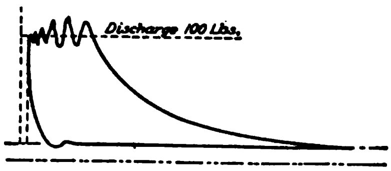


Fig. 8. Single-Stage Air-Compressor Indicator Card,

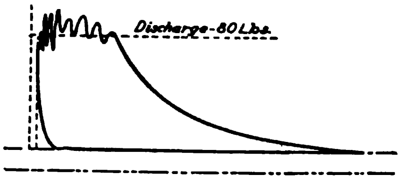


Fig. 9. Single-Stage Air-Compressor Indicator Card.

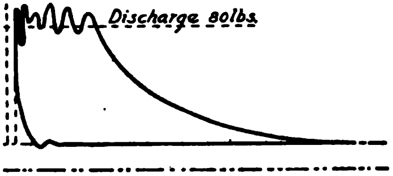


Fig. 10. Single-Stage Air-Compressor Indicator Card.

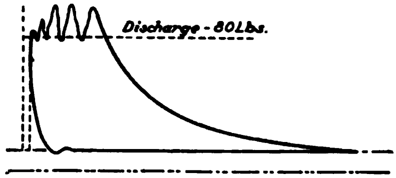


Fig. 11. Single-Stage Air-Compressor Indicator Card.

Fig. 7 is from another 26 by 30-inch single-stage air-cylinder, the terminal pressure being again 61 lbs. gauge, and speed 88 R.P.M. In fact, this card was taken especially for comparison with the card shown in Fig. 6, to show the result of some improvements in the second machine, which was, however, in general type, identical with the first. The operating conditions were therefore made the same in the two cases. It will be noted that the intake line coincides with the atmospheric line for almost the entire stroke; the slight drop in the intake line at beginning of admission suggests the failure of the inlet valve to open fully and instantly, but is really caused by the spring of the indicator; and the slight rise in pressure immediately following this initial drop is due to the reaction of the indicator spring. In this card the volumetric efficiency is 93 per cent—an excellent result for these conditions. The I. H. P. and compression efficiency are not given.

Fig. 8 is another card from this same machine, running, in this case, at 100 R.P.M., and compressing to 100 lbs. gauge.

Figs. 9, 10, and 11 are cards from the same machine with 26 by 30-inch air cylinder at 80 lbs. gauge pressure, and at speeds of 60, 80, and 100 R.P.M. The volumetric efficiency remains practically at 93 per cent at all these speeds, evidencing a very good performance.

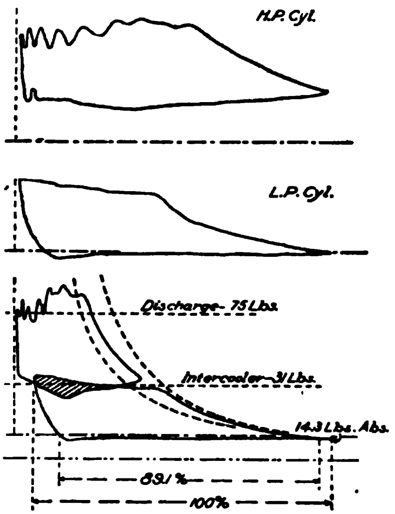


**Fig. 12.** Separate and Combined Indicator Cards from a Tandem Compound Two-Stage Compressor.

Fig. 12 shows the cards from the high- and low-pressure cylinders of a tandem compound compressor of 28 inches stroke and air cylinders 28 and 17½ inches in diameter. The combined card is also shown. The intercooler pressure was 31 lbs.; and the final pressure, 75 lbs. gauge. Volumetric efficiency is 89.1 per cent; I.H.P. of low-pressure cylinder, 129; I.H.P. of high-pressure cylinder, 108; total I.H.P., 237. The efficiency, compared with perfect isothermal compression in two stages, is 80.9 per cent. A considerable vacuum is shown at the beginning of the intake line, which reduces toward the end of the stroke. This would suggest that the inlet valves are not working properly. The re-expansion line indicates a large clearance space in the low-pressure cylinder. The upward slanting low-pressure discharge line suggests a restricted discharge area, or too early closing of the discharge valves. The *peak* on the high-pressure discharge line is rather higher than would ordinarily be considered desirable. The shaded area in the combined card represents work which has been done twice, once in the low-pressure cylinder and later in the high-pressure cylinder. It may be due to a restricted low-pressure discharge, an insufficient high-pressure intake, or an inadequate cross-section in the intercooler.

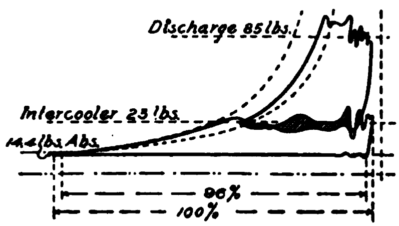


Fig. 13. Combined Cards from a Tandem Compound Two-Stage Compressor.

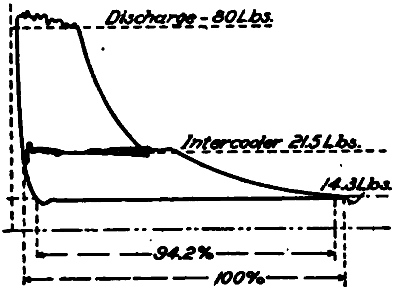


Fig. 14. Combined Cards from a Cross-Compound Two-Stage Compressor.

Fig. 13 presents the separate and combined cards from the air cylinders of another tandem compound machine of a still different type, the size being 20 inches and 13 inches by 12 inches stroke. The speed was 100 R.P.M., and the terminal pressure 85 lbs. gauge, with atmosphere at 14.7 lbs. absolute. The volumetric efficiency is 96 per cent — an exceptionally good showing. The I.H.P. of the low-pressure cylinder is 44; and of the high-pressure cylinder, 52.25; total I.H.P., 96.25. The efficiency of compression, compared with perfect isothermal two-stage, is 91.25 per cent. Taken as a whole, this is a card rather above the average from this type of machine, although the shaded area in the combined card suggests possibilities of improvement in the intercooler and in the valve action.

Fig. 14 gives separate and combined cards from a double cross-compound compressor with an air end 34 inches and 24 inches by 24 inches stroke, running at 94 R.P.M., and compressing at sea-level to 80 lbs. gauge. The total I.H.P. is 324; and the compression efficiency compared with two-stage isothermal, is 91.7 per cent. Volumetric efficiency is 94.2 per cent. Taken as a whole, this is a splendid record of performance, though some slight defects may be noted in the details of the cards.

## TABLE VII

### Horse-Power Developed in Compressing One Cubic Foot of Free Air from Atmospheric Pressure (14.7 Pounds) to Various Gauge Pressures

INITIAL TEMPERATURE OF THE AIR IN EACH CYLINDER TAKEN AS 60° F.
(Jacket Cooling Not Considered)

| GAUGE PRESSURE | ISOTHERMAL COMPRESSION | ADIABATIC COMPRESSION | | | |
|---|---|---|---|---|---|
| | | One-Stage | Two-Stage | Three-Stage | Four-Stage |
| 10 | .0332 | .0358 | | | |
| 20 | .0551 | .0623 | | | |
| 30 | .0713 | .0842 | | | |
| 40 | .0842 | .1026 | | | |
| 50 | .0950 | .1187 | | | |
| 60 | .1042 | .1331 | | | |
| 70 | .1122 | .1465 | .128 | .122 | .119 |
| 80 | .1194 | .1585 | .137 | .131 | .127 |
| 90 | .1258 | .1695 | .146 | .139 | .135 |
| 100 | .1317 | .1800 | .154 | .146 | .142 |
| 125 | .1443 | .2036 | .171 | .161 | .157 |
| 150 | .1549 | .2244 | .186 | .174 | .169 |
| 200 | .1719 | .2600 | .210 | .196 | .190 |
| 300 | .1964 | .3164 | .247 | .229 | .220 |
| 400 | .2141 | .3613 | .276 | .253 | .242 |
| 500 | .2279 | .3889 | .299 | .272 | .260 |
| 600 | .2393 | .4318 | .318 | .288 | .275 |
| 700 | .2489 | .4608 | .335 | .302 | .289 |
| 800 | .2573 | .4873 | .349 | .314 | .299 |
| 900 | .2649 | .5114 | .363 | .325 | .310 |
| 1,000 | .2720 | .5337 | .375 | .335 | .318 |
| 1,200 | .2829 | .5742 | .397 | .353 | .333 |
| 1,400 | .2924 | .6102 | .414 | .368 | .347 |
| 1,600 | .3012 | .6427 | .432 | .381 | .359 |
| 1,800 | .3087 | .6724 | .447 | .393 | .369 |
| 2,000 | .3154 | .7003 | .460 | .403 | .379 |

NOTE—The above values are for sea-level conditions only. For the power developed at altitudes, see Table IX (p. 30).

**Power Required to Compress Air.** Table VII, compiled by F. M. Hitchcock, lists the power required or developed per cubic foot of free air in compressing from atmospheric pressure of 14.7 lbs. absolute to the gauge pressures listed in the first column. The second column gives the isothermal or theoretical minimum horse-power required, never equaled in practice. In the next four columns are given the horse-powers required for adiabatic compression in one, two, three, and four stages. It will be noted that this table does not consider stage compression for pressures below 70 lbs. gauge; and a comparison of the figures in these last four columns, for any given pressure, throws an interesting light on the way in which the possibilities of gain by compounding in various degrees grow larger as pressures increase.

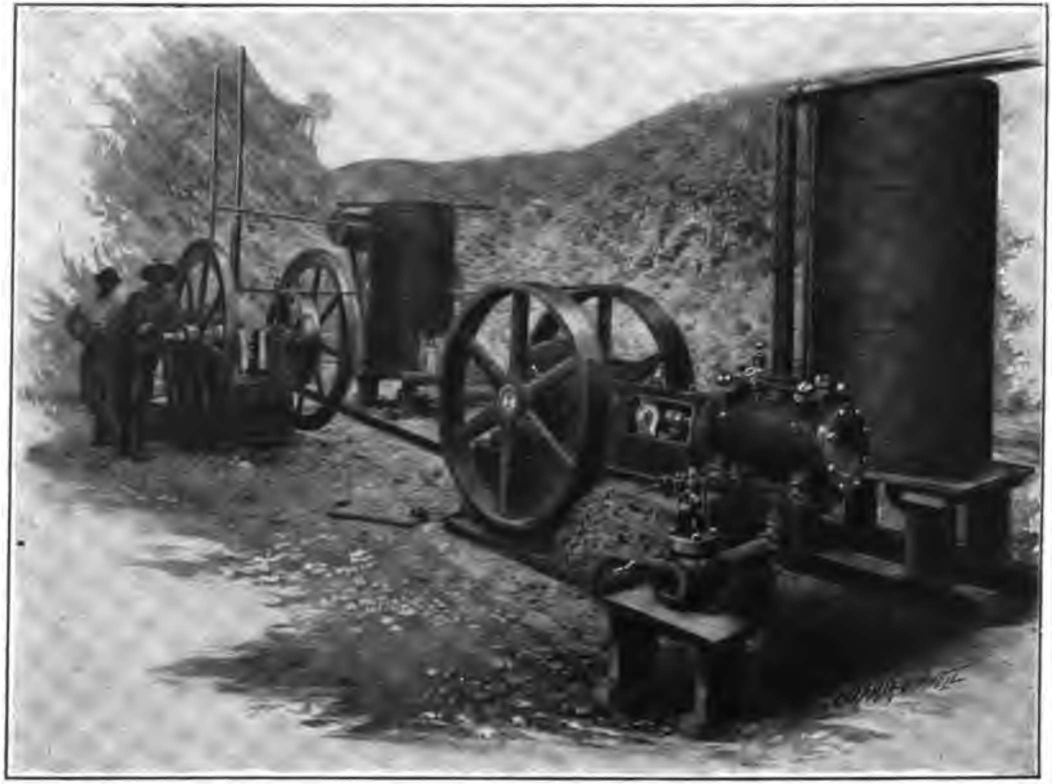
It is to be noted, also, that this table is based on sea-level conditions, and assumes that, whether in single- or multi-stage compression, the initial temperature of the air in each cylinder is 60° F. In other words, in the compound machines, the intercooler is supposed to reduce the temperature of the air to the initial temperature in the low-pressure cylinder. These figures are theoretical only, and no allowance has been made for the usual losses found in actual compressor operation. Increasing these figures by 10 or 15 per cent would liberally cover such losses and serve as a safe approximation to the conditions of everyday work.

### TABLE VIII

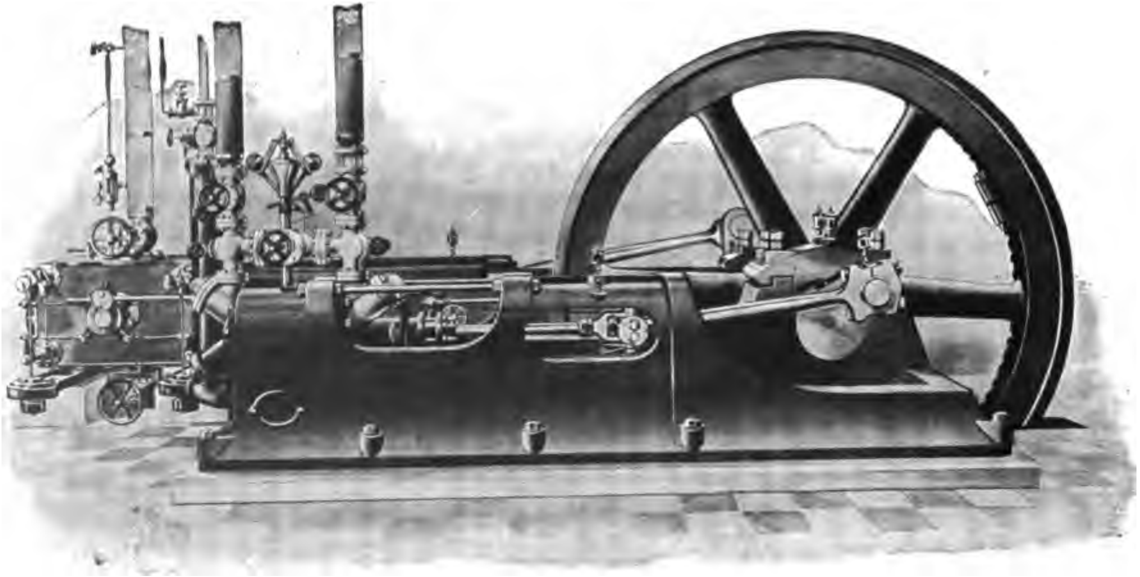
**Multipliers for Determining the Volume of Free Air at Various Altitudes, which, when Compressed to Various Pressures, is Equivalent in Effect to a Given Volume of Free Air at Sea-Level**

| Altitude in Feet | Barometric Pressure | | Multiplier | | | | |
|---|---|---|---|---|---|---|---|
| | | | Gauge Pressure (Pounds) | | | | |
| | Inches of Mercury | Pounds per Square Inch | 60 | 80 | 100 | 125 | 150 |
| 0 | 30.00 | 14.75 | 1.000 | 1.000 | 1.000 | 1.000 | 1.000 |
| 1,000 | 28.88 | 14.20 | 1.032 | 1.033 | 1.034 | 1.035 | 1.036 |
| 2,000 | 27.80 | 13.67 | 1.064 | 1.066 | 1.068 | 1.071 | 1.072 |
| 3,000 | 26.76 | 13.16 | 1.097 | 1.102 | 1.105 | 1.107 | 1.109 |
| 4,000 | 25.76 | 12.67 | 1.132 | 1.139 | 1.142 | 1.147 | 1.149 |
| 5,000 | 24.79 | 12.20 | 1.168 | 1.178 | 1.182 | 1.187 | 1.190 |
| 6,000 | 23.86 | 11.73 | 1.206 | 1.218 | 1.224 | 1.231 | 1.234 |
| 7,000 | 22.97 | 11.30 | 1.245 | 1.258 | 1.267 | 1.274 | 1.278 |
| 8,000 | 22.11 | 10.87 | 1.287 | 1.300 | 1.310 | 1.319 | 1.326 |
| 9,000 | 21.29 | 10.46 | 1.329 | 1.346 | 1.356 | 1.366 | 1.374 |
| 10,000 | 20.49 | 10.07 | 1.373 | 1.394 | 1.404 | 1.416 | 1.424 |

**Air-Compression at Altitudes.** As altitude above sea-level increases, the atmospheric pressure diminishes, and the density of the air is correspondingly reduced. The first three columns of Table VIII, compiled by F. M. Hitchcock, show the relation between altitude and atmospheric pressure. This phenomenon of progressive rarification of the air as heights increase, has an important bearing upon air-compression at altitudes.



SULLIVAN AIR COMPRESSOR, DRIVEN BY 30-H. P. GASOLINE ENGINE



TANGYE FRAME HORIZONTAL LINDE COMPRESSOR.
Fred W. Wolf Company.

Air-compressors are rated in cubic feet of free air per minute, but no atmospheric pressure is usually specified. This capacity is ordinarily given as the actual piston displacement of the machine; but this is not correct, since there are always certain losses reducing the volumetric efficiency below unity. When, however, a given amount of work is to be done by the compressor, the atmospheric pressure is an important factor in the problem. For instance, a 2-inch rock-drill is usually rated at an air consumption of 50 cubic feet of free air per minute, compressed to 60 lbs. gauge. If it requires this volume of free air, and the equivalent volume of compressed air at 60 lbs. pressure, at sea-level, when operating at (say) 9,000 feet altitude and supplied with 50 cubic feet of free air per minute compressed to 60 lbs. pressure, it will fall far short of its full rated performance. This is because the equivalent volume at 60 lbs. pressure of 50 cubic feet of free air at sea-level, or 14.7 lbs. absolute pressure, is considerably more than the equivalent volume at 60 lbs. gauge of 50 cubic feet of free air at 9,000 feet altitude, where the atmospheric pressure is only 10.46 lbs. absolute. To provide for this, therefore, a larger volume of free air must be compressed at altitudes than at sea-level, to get a given amount of work out of a machine having a certain rating in cubic feet of compressed air at sea-level.

Columns 4 to 8 of Table VIII give the multipliers for determining what volume of free air at various altitudes and atmospheric pressures it will be necessary to compress in order to produce the equivalent in air under pressure of a given volume of sea-level free air. For example, ten 3¼-inch rock-drills, each rated at 100 cubic feet of free sea-level air at 60 lbs. gauge pressure, are to be operated simultaneously at 6,000 feet altitude; what volume of free air must be supplied them to give rated performance at this altitude? At sea-level the ten drills would require 1,000 cubic feet of free air per minute. The multiplier, from the table, for 6,000 feet altitude and 60 lbs. pressure, is 1.206. The required volume of free air at the higher altitude is therefore 1,000 times 1.206, or 1,206 cubic feet of free air per minute compressed to 60 lbs. gauge. It is evident from this example and from the accompanying discussion, that an increase in altitude is equivalent to a reduction in the volumetric efficiency of the compressor.

*Horse-Power for Compression at Altitudes.* The power required to compress a given volume of free air to a given pressure, diminishes as the altitude increases. This will be apparent upon a little consideration of the compressed-air indicator card. It will be remembered that such a card is made up of two areas representing two distinct stages of work—*first,* compression to the stated pressure; *second,* discharge at that pressure. As the density of the free air (or the atmospheric pressure) diminishes with increased altitude or otherwise, it is evident that the piston will have to travel further into the cylinder before the required terminal pressure is reached and discharge begins. This will increase the area of compression work and reduce the area showing discharge work. But the latter decreases more rapidly than the former increases; hence the total power represented by the combined areas will be less.

### TABLE IX

### Horse-Power Developed, with Allowance for Usual Losses, in Compressing One Cubic Foot of Free Air at Various Altitudes from Atmospheric to Various Pressures

Initial Temperature of the Air in Each Cylinder Taken as 60° F.
(Jacket Cooling Not Considered)

| Altitude in Feet | Horse-Power Developed | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | Simple Compression | | | Two-Stage Compression | | | | |
| | Gauge Pressure (Pounds) | | | Gauge Pressure (Pounds) | | | | |
| | 60 | 80 | 100 | 60 | 80 | 100 | 125 | 150 |
| 0 | .1533 | .1824 | .2075 | .1354 | .1580 | .1765 | .1964 | .2138 |
| 1,000 | .1511 | .1795 | .2040 | .1332 | .1553 | .1734 | .1926 | .2093 |
| 2,000 | .1489 | .1766 | .2006 | .1310 | .1524 | .1700 | .1887 | .2048 |
| 3,000 | .1469 | .1739 | .1971 | .1286 | .1493 | .1666 | .1848 | .2003 |
| 4,000 | .1448 | .1712 | .1939 | .1263 | .1464 | .1635 | .1810 | .1963 |
| 5,000 | .1425 | .1685 | .1906 | .1241 | .1438 | .1600 | .1772 | .1921 |
| 6,000 | .1402 | .1656 | .1872 | .1218 | .1409 | .1566 | .1737 | .1879 |
| 7,000 | .1379 | .1628 | .1839 | .1197 | .1383 | .1536 | .1700 | .1838 |
| 8,000 | .1358 | .1600 | .1807 | .1173 | .1358 | .1504 | .1662 | .1797 |
| 9,000 | .1337 | .1572 | .1774 | .1151 | .1329 | .1473 | .1627 | .1758 |
| 10,000 | .1316 | .1547 | .1743 | .1132 | .1303 | .1442 | .1592 | .1717 |

Table IX, also compiled by F. M. Hitchcock, gives the horsepower required to compress a cubic foot of free air at various altitudes to various pressures, by one- and two-stage compression. In the figures given, a fair allowance has been made for the ordinary losses, so that these values may be taken as representing average conditions of practice. This table, in connection with Table VIII, affords the solution of a very common problem. For instance, what power will be required to secure by two-stage compression at 9,000 feet altitude, the equivalent of 1,000 cubic feet of free air at sea-level at 100 lbs. gauge pressure? From Table VIII, the multiplier for determining the volume is found to be 1.356; from Table IX, the horse-power required per cubic foot of free air compressed at this altitude to this pressure is found to be .1473. The required result, therefore, is 1,000 × 1.356 × .1473, or 199.73 horse-power.

Tables VII, VIII, and IX are here used by permission of the Ingersoll-Rand Company.

### TABLE X

**Relation of Volumetric Efficiencies and Horse-Power Required in Air-Compression at Various Altitudes**

| ALTITUDE IN FEET | ATMOSPHERIC PRESSURE, LBS. ABS. | VOLUMETRIC EFFICIENCY | LOSS OF VOLUMETRIC EFFICIENCY | PER CENT POWER REQUIRED | PER CENT REDUCTION IN POWER REQUIRED |
|---|---|---|---|---|---|
| Sea-Level | 14.75 | 1.00 | .00 | 1.000 | .000 |
| 1,000 | 14.20 | .97 | .03 | .982 | .018 |
| 2,000 | 13.67 | .93 | .07 | .965 | .035 |
| 3,000 | 13.16 | .90 | .10 | .948 | .052 |
| 4,000 | 12.67 | .87 | .13 | .931 | .069 |
| 5,000 | 12.20 | .84 | .16 | .915 | .085 |
| 6,000 | 11.73 | .81 | .19 | .899 | .101 |
| 7,000 | 11.30 | .78 | .22 | .884 | .116 |
| 8,000 | 10.87 | .76 | .24 | .869 | .131 |
| 9,000 | 10.46 | .73 | .27 | .854 | .146 |
| 10,000 | 10.07 | .70 | .30 | .839 | .161 |
| 11,000 | 9.70 | .68 | .32 | .824 | .176 |
| 12,000 | 9.34 | .65 | .35 | .809 | .191 |
| 13,000 | 8.98 | .63 | .37 | .794 | .206 |
| 14,000 | 8.65 | .60 | .40 | .779 | .221 |
| 15,000 | 8.32 | .58 | .42 | .765 | .235 |

Table X shows the relation of volumetric efficiencies and horse-power required in air-compression at various altitudes. It will be noted that the volumetric efficiency decreases more rapidly than the required power.

**Unavoidable Losses in Air-Compression.** From the preceding discussion of the general theory and phenomena of air-compression, the following concrete deductions may be made as to the losses which are unavoidable in practical air-compression work—losses which can be kept down to the minimum by correct design and construction, but which can never be entirely eliminated.

The *first loss* is that due to friction in the mechanical structure which transmits the power from the driving to the compressing element.

The *second loss* is that arising from the heating of the air during admission to the cylinder, resulting in a reduction in its density and a diminished volumetric efficiency of the cylinder.

The *third loss* arises from the heat produced by compression, and its effect upon the air during compression, calling for more power.

The *fourth loss* comes from the reduced volumetric efficiency of the cylinder due to the fact that the intake air is seldom at atmospheric pressure, owing to friction in the inlet valves and passages.

The *fifth loss* is also one of volumetric efficiency, due to the effect of the clearance air in reducing the effective length of the admission stroke.

It is not possible to give even average percentages for any of these losses, which will occur in different combinations depending on the mechanical details, the general type, the speed, and the capacity of the compressor. After all these losses have been deducted, the *net efficiency* of the machine remains; and of this, Mr. Frank Richards, in his work on "Compressed Air," says: "It is safe to say that the ultimate efficiency never goes as high as 80 per cent, while it often goes below 60 per cent."

## CLASSIFICATION OF COMPRESSOR TYPES

Any air-compressor may be considered as made up of three primary parts—the *driving element;* the *compressing element;* and the mechanical structure uniting these two, which may be designated as the *power-transmitting element.* It is not always possible to draw the line clearly between these several elements, but this subdivision can always be made with more or less exactness.

The first classification of compressor types will be based upon the motive power or method of drive—the driving element. This division affords two basic types—*Steam-Driven* and *Power-Driven* compressors The former type includes all those compressors in which a steam cylinder (or cylinders) is an integral part of the machine. The latter includes all those compressors where power is applied from a source outside of the machine, such as a driving shaft, an electric motor, or a water wheel.

Power-driven compressors may be still further classified by the means employed for applying the power, into *Belt-Driven, Rope-Driven, Chain-Driven, Gear-Driven,* and *Direct Shaft Connected* (with motor or water wheel).

Another classification is based upon the direction in which the reciprocating parts of the machine work; and its divisions include *Vertical Compressors* and *Horizontal Compressors.*

The third general classification refers to the relative position of the driving and compressing elements. When a steam-driven compressor has all its steam and air cylinders arranged on an axial line with their pistons on one long rod, the type is designated as a *Straight-Line Compressor.* When the compressor consists of two distinct elements, each comprising a driving and a compressing element, and these two elements are arranged side by side with a common shaft, the type is known as a *Duplex Compressor.* These two latter types, of course, originated in the steam-driven machine; but the general design has been carried on into the power-driven class, so that there are straight-line and duplex power-driven compressors as well.

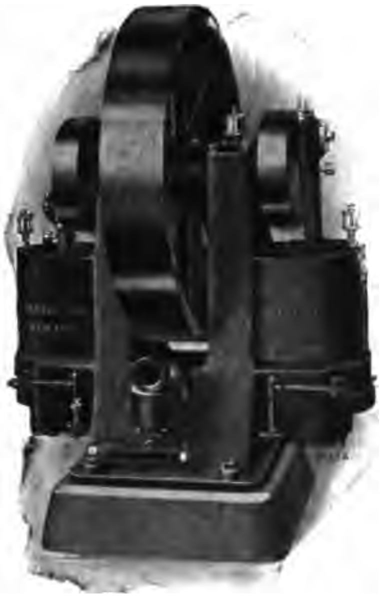


Fig. 15. "Imperial Type XI-1" Vertical Duplex Power-Driven Air- Compressor. Ingersoll-Rand Co., New York City.

The fourth general classification has to do entirely with the compressing end. A compressor in which all the process of compression is carried on in one cylinder, is known as a *Single-Stage Compressor.* A duplex compressor having two air-cylinders, in each of which compression is carried from initial pressure to terminal or discharge pressure, is called a *Duplex Single-Stage Compressor.* When the

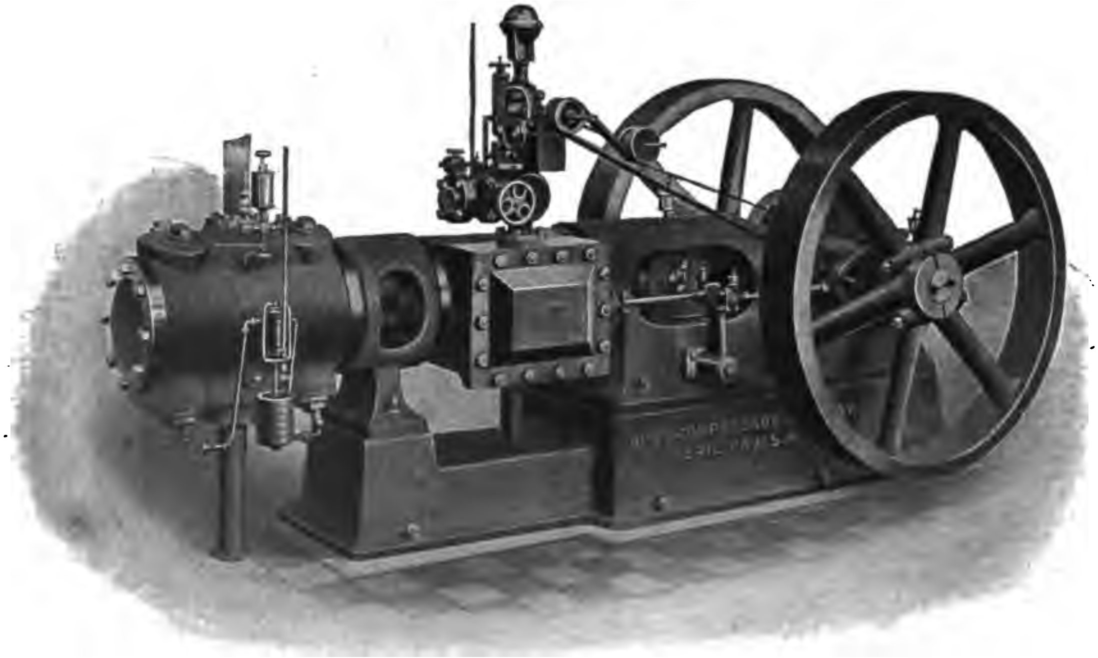


Fig. 16. Horizontal Straight-Line Simple Steam Single-Stage Compressor, Class BS.
Bury Compressor Company, Erie, Pa.

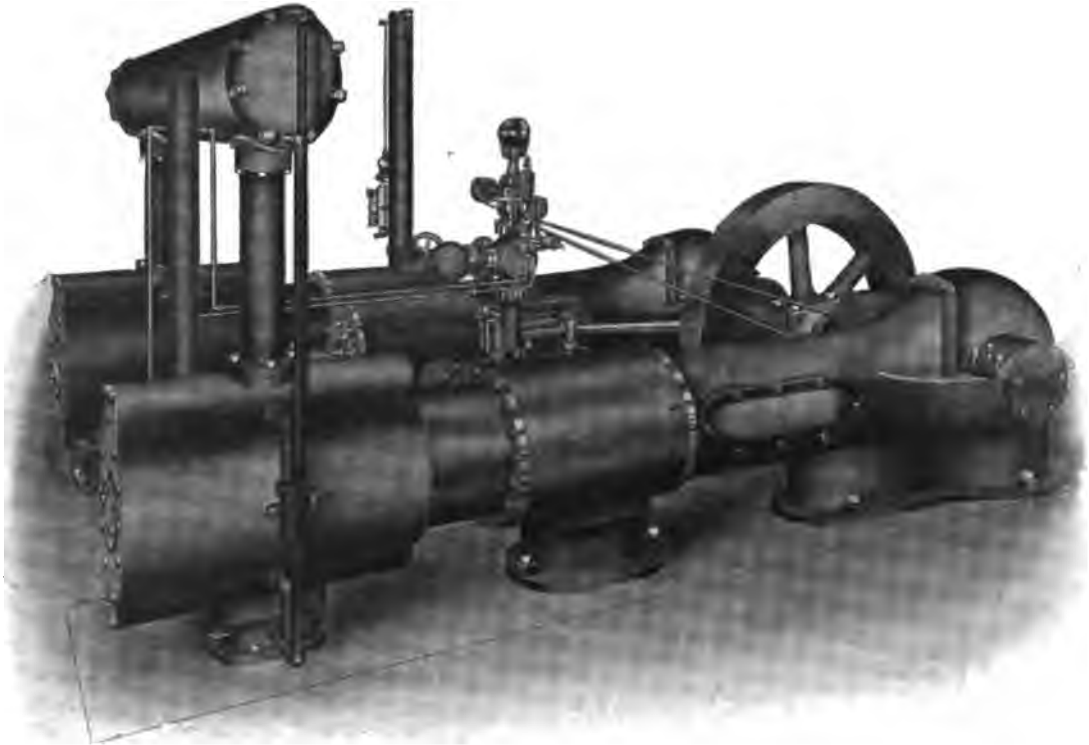


Fig. 17. Class NT-32 Double Cross-Compound Steam Driven Air-Compressor. J. Geo. Leyner Engineering Works Company, Littleton, Col.

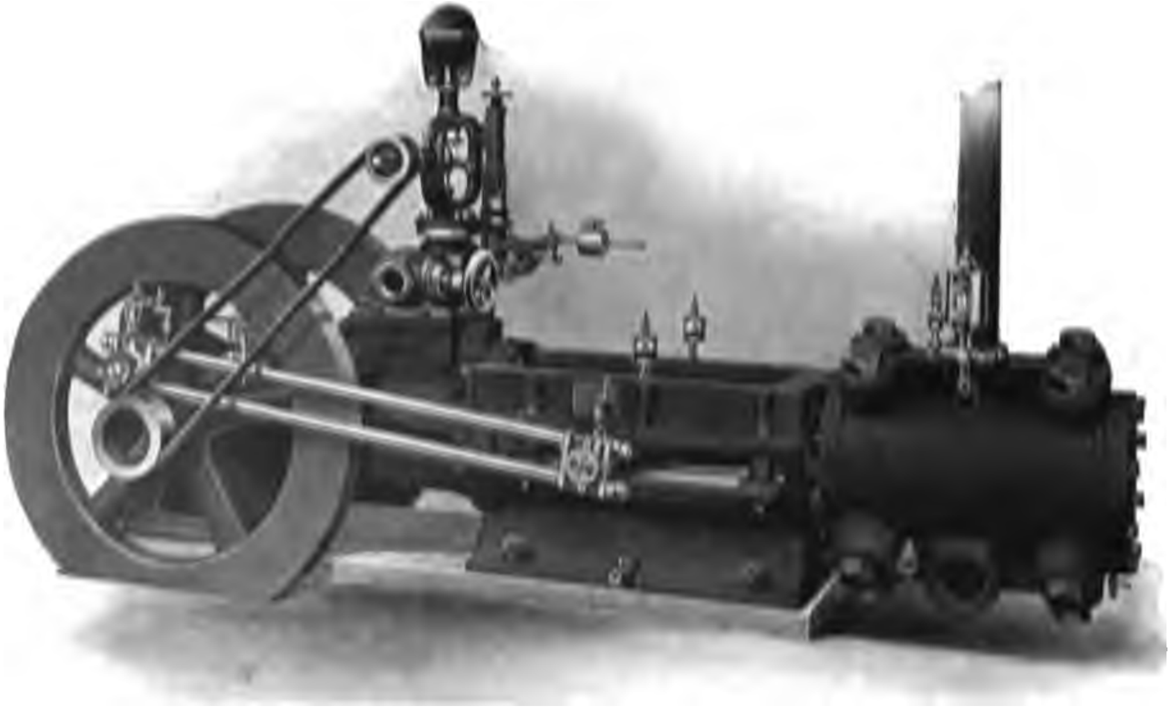


Fig. 18. Class WA 8 Straight Line Simple Steam Single Stage Air Compressor. Sullivan Machinery Co., Chicago, Ill.

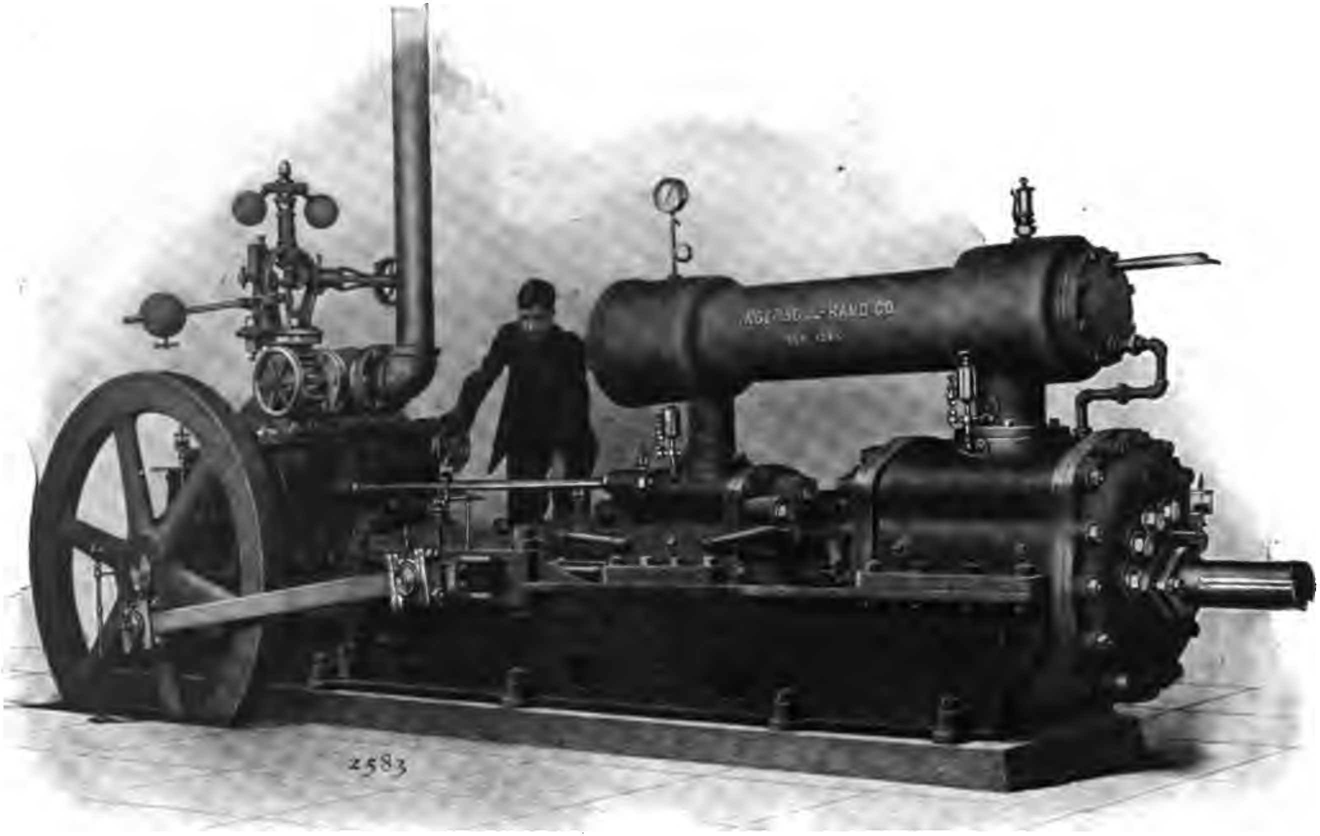


Fig. 19. Class A-2 Simple Steam Tandem Compound Straight-Line Air-Compressor.
A typical 18-inch stroke machine with tandem compound air cylinders and horizontal overhead intercooler.
Ingersoll-Rand Co., New York City.

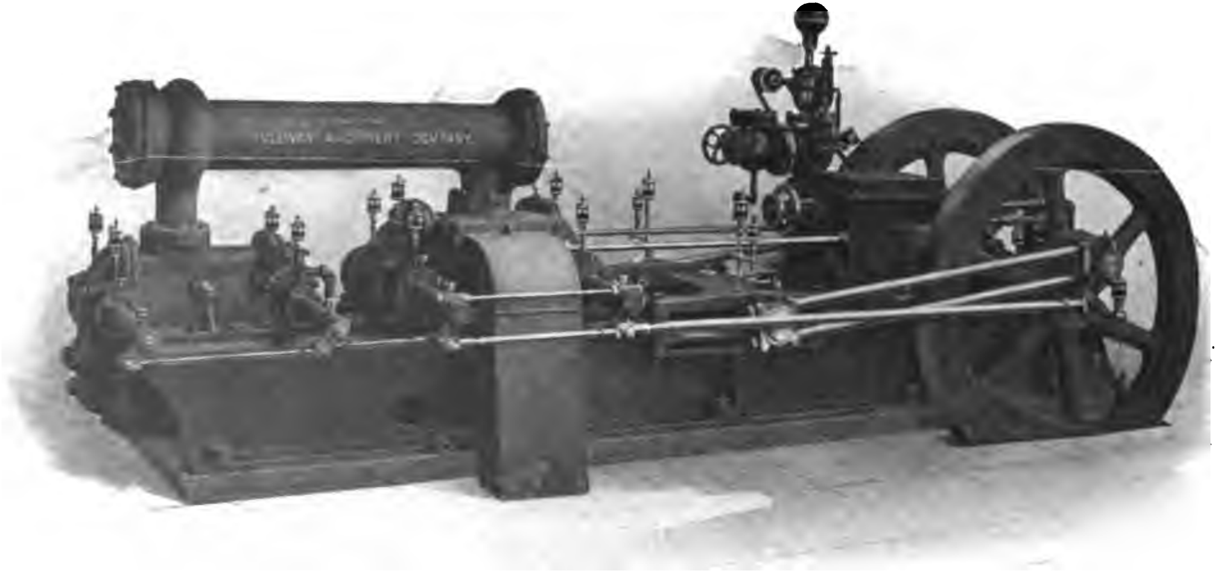


Fig. 20. Class WB-2 Straight-Line Simple Steam Tandem Compound Air Compressor. Sullivan Machinery Co., Chicago, Ill

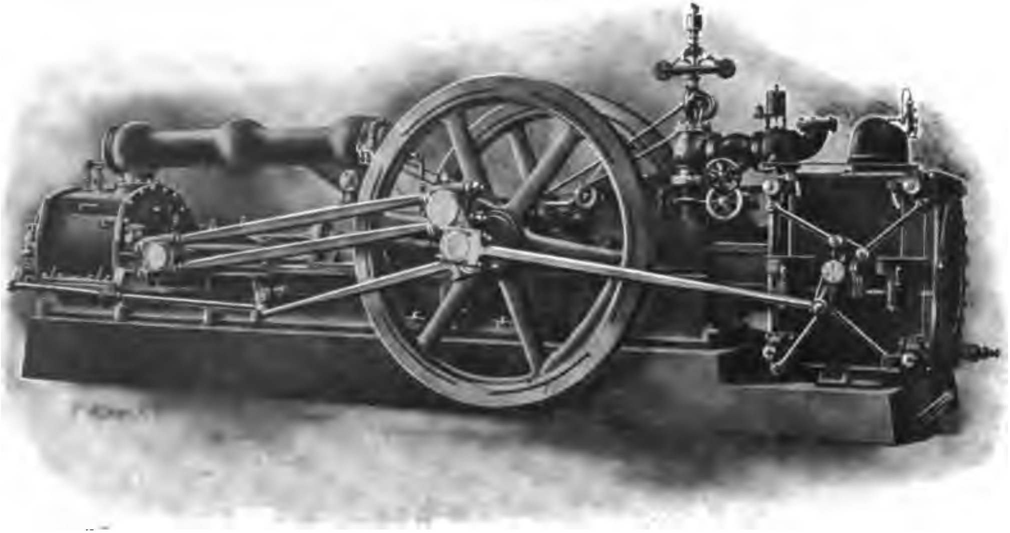


Fig. 21. Tandem Double-Compound Straight-Line Air-Compressor.
Norwalk Iron Works Company, South Norwalk, Conn.

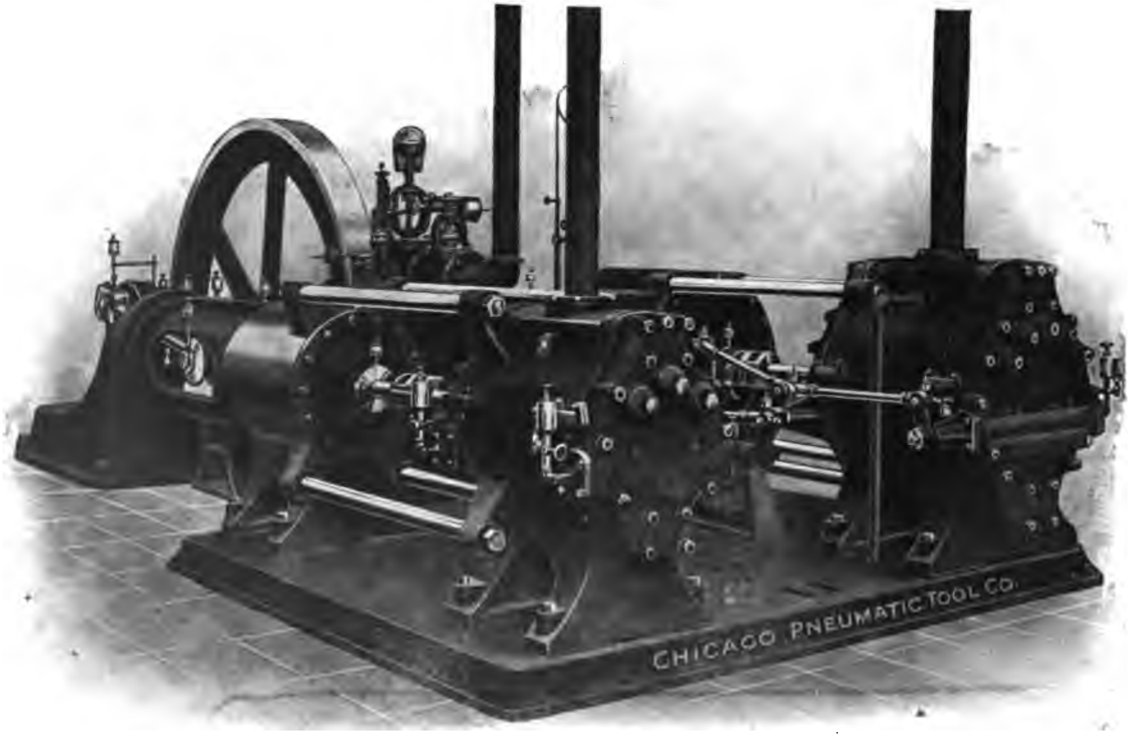


Fig. 22. Franklin Corliss Double Cross-Compound Steam-Driven Air-Compressor.
Compound steam cylinders and two-stage air cylinders. (Separate intercooler not shown.)
Chicago Pneumatic Tool Company, Chicago, Ill.

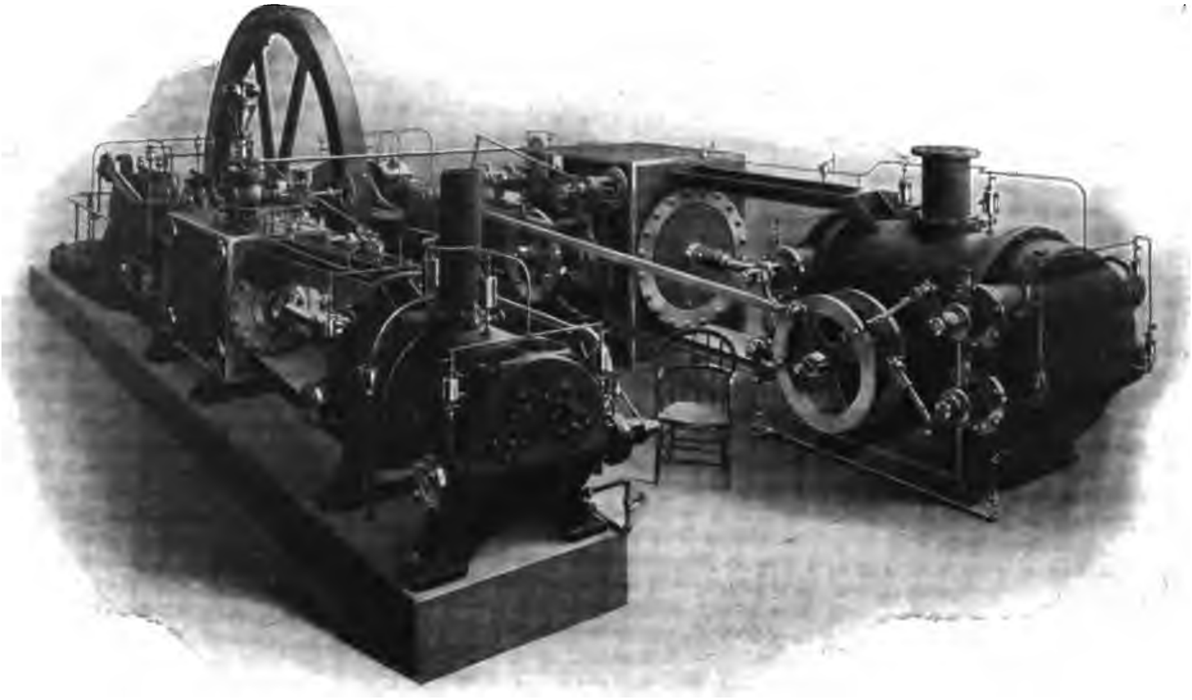


**Fig. 23.** Class WX Corliss Double Cross-Compound Steam-Driven Air Compressor. Sullivan Machinery Co., Chicago, Ill.

compression process is divided between two cylinders, in the first of which a certain pressure is reached, and in the second of which the pressure is carried from the terminal pressure of the first cylinder to final discharge pressure, the machine is called a *Two-Stage Compressor.* Where three degrees of compression in three cylinders are used, the machine is a *Three-Stage Compressor*; and with four degrees or stages of compression in four cylinders, a *Four-Stage Compressor.* The term *compound,* as applied to air cylinders, usually implies a two-stage machine, while *multi-stage* is usually applied to three and four stages.

A straight-line compressor with simple steam cylinder and a single-stage air cylinder, is spoken of as a *Simple Steam Single-Stage Straight-Line* or a *Simple Tandem Straight-Line.* When a two-stage or three-stage air end is used in a straight-line machine in connection with a simple air cylinder, the machine is designated as a *Simple Steam Tandem Compound* or a *Simple Steam Tandem Three-Stage.* When the steam cylinders are also compounded, the machine becomes a *Tandem Double-Compound* or a *Tandem Double-Compound Three-Stage.*

A duplex compressor with two simple steam cylinders and two single-stage air-cylinders is a *Simple Duplex*; with compound steam cylinders and two single-stage air cylinders, a *Cross-Compound Duplex Air*; with compound steam cylinders and two-stage air cylinders, a *Double Cross-Compound*; with compound steam and multi-stage air end, a *Double Cross-Compound Three-* or *Four-Stage*; with simple steam and compounded or multi-stage air cylinders, a *Duplex Steam Cross-Compound* or a *Duplex Steam Cross-Compound Three-* or *Four-Stage.*

While different methods of designation are found to hold among different manufacturers, those just given are based on rational grounds, and will be adopted in this treatise for the sake of uniformity.

There are, of course, compressor types which do not conform exactly to any of the classifications here given; but an analysis of their designs, with proper allowance for certain features more or less special in character, will place them in some of the classes here defined. There is a growing tendency toward a standardization of types among all the large builders; and in the present work on Compressed Air, only the best modern practice will be considered.

## STEAM-DRIVEN COMPRESSORS

The ordinary steam-driven compressor is simply a steam engine with one or more air cylinders directly coupled on the extended piston-rod. With this understanding, it will be unnecessary here to enter into any detailed discussion of the steam end of the air-compressor.

It will be enough to state that the smaller steam-driven compressors generally use a plain steam valve—either of D or piston type—with a fixed point of cut-off adjusted for the best steam economy under average conditions. Following this first and simplest class of steam ends, comes the class in which adjustable cut-off steam valves are used, either of the flat or piston type, balanced or unbalanced as the case may be. A third general class is that in which Corliss steam valves, more or less modified, are used; and these are usually found on the larger sizes of compressors, where the very best steam economy is sought. These Corliss valves are either of the "drop-release" pattern, or of the fixed type with no release. Corliss steam valves, to realize their best economy, should be used with a condensing apparatus.

**Distinction between Steam-Engine and Compressor Practice.** There is one vital point of difference between steam-engine and air-compressor practice which is to be borne in mind throughout all the discussion that follows. While the steam engine for ordinary power service is designed to maintain a constant speed irrespective of load, the steam-driven air-compressor must run at varying speed, since its output of air is proportional to its piston displacement, and the latter in turn is proportional to the speed. This means that while the steam engine in ordinary power service has a variable load per stroke, in air-compressor practice the steam cylinder has a *constant load per stroke,* except for the small discrepancy due to varying friction and compressive efficiencies at varying speeds.

**Compound Steam Cylinders on Air-Compressors.** It is clearly understood that the fundamental object of compounding steam cylinders is to secure a higher economy by the more complete expansion of the steam—that is, to make more complete the abstraction of the heat energy of the steam by increasing the difference between the initial and terminal temperatures.

Another advantage of compounded steam cylinders, which may be referred to later, is that the maximum pressure in compound cylinders—and therefore the maximum load on valves, valve gears, etc. —is less than in an equivalent simple steam cylinder. This means lower friction losses, lower terminal pressures on bearings, etc., and a general improvement in the operating conditions.

The first essential of successful steam compounding is in securing the proper ratio of volumes in high- and low-pressure cylinders, so that when a given quantity of steam (determined by the cut-off on the high-pressure cylinder) is admitted and expanded through the two cylinders, its terminal or exhaust pressure shall be such that there shall be no loss by condensation due to over-expansion. This means that for a given steam pressure, for a given exhaust or back pressure, and for a given cylinder ratio, there is one and only one proper point of cut-off for both cylinders, to give the best economy. In the relation just expressed, the first and third elements must be considered as constant, leaving the latter element as the variable to be adjusted to meet varying conditions.

Now, where a constant speed must be maintained regardless of load, this change of speed must evidently be produced by a change of cut-off; and this immediately destroys the proper adjustment, and the best economy is at once sacrificed. Constant-speed practice with compound engines must at the best be a compromise based on varying load conditions.

But in the air-compressor there is a constant load per stroke (ignoring friction for the present) so that the cut-off remains practically constant at all speeds, and the correct ratio of expansion is maintained over the full load range. The word *practically* is used above with deliberate intent, since there is bound to be a very slight departure from perfect conditions to produce the requisite change of speed under changing load, as will be seen in a subsequent section on Regulation. But enough has been said here to demonstrate that the air-compressor offers the ideal opportunity for securing in highest degree the advantages of steam compounding.

**Steam Pressures for Compounding.** Within practical limits, the higher the initial steam pressure, the greater are the advantages of compounded steam cylinders. Compressors have been built with triple- and quadruple-expansion cylinders using steam pressures as high as 250 lbs. These machines, however, are so rare that they need not enter into a discussion of general compressor practice; and the most that the average practical man may expect to encounter is a compound or double-expansion compressor. The peculiar operating conditions of the air-compressor, just explained, justify the adoption of compounded steam ends on pressures lower than would be advisable for compounding in ordinary steam-engine practice. The largest builders of air and gas compressors to-day are recommending compounded steam ends for 80 lbs. boiler pressure where the machine is to be run condensing, and for 90 lbs. boiler pressure running non-condensing. Above these minima the pressures may run up to the limit of everyday practice, which may be as high as 125 or even 150 lbs. gauge.

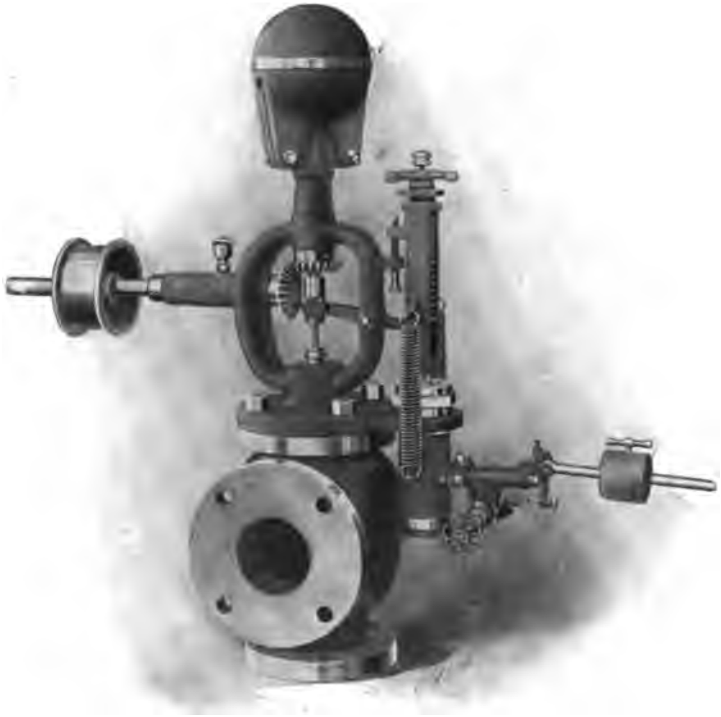


Fig. 24. Combined Speed and Pressure Regulator for Steam-Driven Compressors. Sullivan Machinery Co., Chicago, Ill.

The three fundamental advantages of compounded steam cylinders are as follows:

Lower fuel consumption and fuel cost, owing to the smaller quantity of water to be evaporated.

Lower cost of boiler feed-water, as a result of the greater power secured per pound of water by the higher expansion.

Lower cost of boiler plant and accessories, because of the reduced amount of water required to be evaporated.

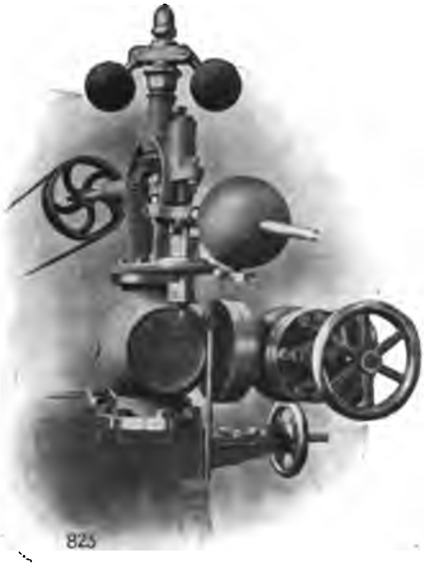


**Fig. 25.** Combination Air Ball Governor and Pressure Regulator for Steam-Driven Compressors.
Ingersoll-Rand Co., New York City.

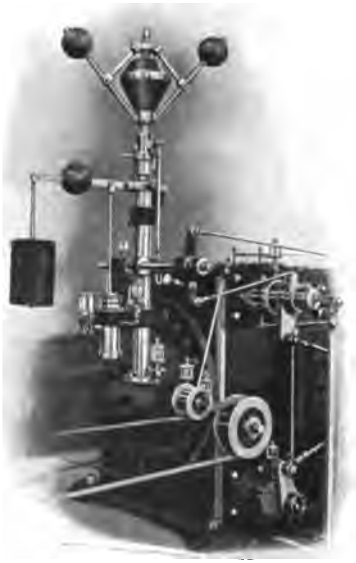


**Fig. 26.** Class A-15 Combined Speed and Pressure Regulator for Corliss Compressors.
Ingersoll-Rand Co., New York City.

Other incidental advantages of steam compounding will be developed later in another connection.

**Regulation of Steam-Driven Compressors.** While the regulating devices or *governors* used on air-compressors vary in detail among the different builders, all may be divided, so far as the principle of operation is concerned, into two classes.

The first class is that applied to compressors with plain or adjustable cut-off valves of flat or piston type. It operates by throttling the steam supply as load diminishes. Devices in this class consist fundamentally of a valve in the steam pipe, which is opened or closed by the action of a piston in a cylinder, this piston being actuated by air pressure from the receiver. The movement of this piston is opposed by weights on a lever or by a spring; and the spring tension or the weights may be adjusted so that the governor valve is full open at any desired normal air pressure. But when pressure exceeds this limit, the tension or weight is overcome, and the valve in the steam supply is closed in a degree corresponding with the amount of excess pressure. This slows down the machine and reduces the volume of air discharged until such time as the normal pressure is reached, when the weights or spring tension again open the governor valve, and full speed is restored. Evidently this partial throttling of the steam supply will result in some wire-drawing of the steam, which is about offset by the resulting superheating of the steam.

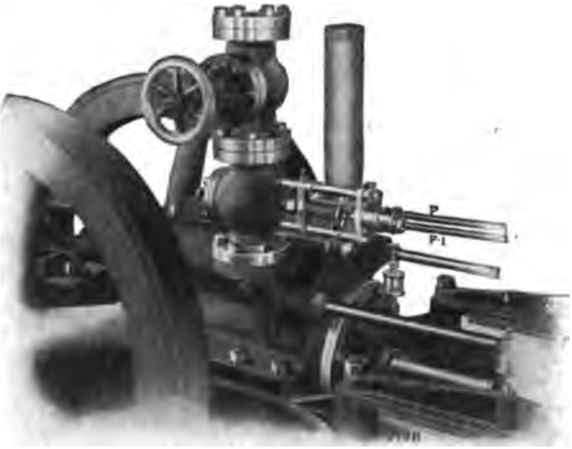


Fig. 27. Sergeant A-14 Steam Regulator.
Automatically regulates speed under varying load by throttling steam supply.
Ingersoll-Rand Co., New York City.

The second class includes governors applied to machines with Corliss steam valves. The mechanism consists of a pressure cylinder and piston with the opposing weights or springs as described in the preceding paragraph; but in this case the movement of the governor, instead of throttling the steam, changes the cut-off of the steam valves, reducing speed under partial load, and restoring it as load increases. The resistance per stroke being the same through-

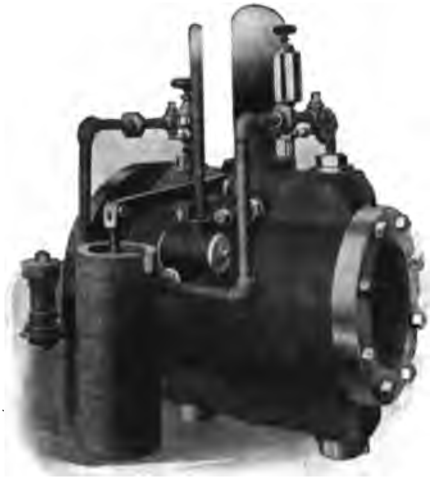


Fig. 28. Sergeant A-28 Air Unloader Used with Steam Regulator Shown in Fig. 27.

out, a very slight change of cut-off, hardly affecting the economy, produces all the speed change necessary.

Both of these classes of governors include also a speed-limiting device, the more common form being the familiar fly-ball arrangement, which throttles the steam supply or greatly shortens the cut-off when speed exceeds a certain limit, as it might in case an air-pipe should break or other accident occur.

There are occasional instances in which governors of one or other of the above classes are used in connection with an *unloader* on the air end, so arranged that, as speed falls off, the load is partially taken from the air end. These are the specialties of individual builders, and as such cannot be elaborated upon here. The two general classes defined cover the general requirements of this paper.

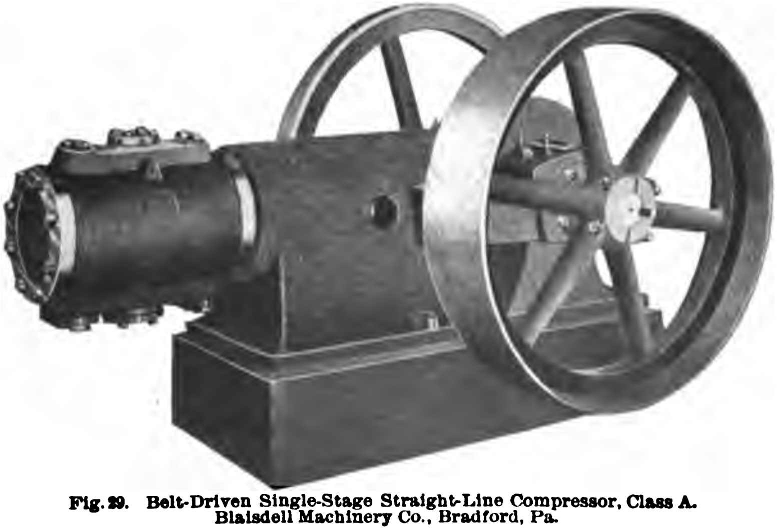


Fig. 29. Belt-Driven Single-Stage Straight-Line Compressor, Class A. Blaisdell Machinery Co., Bradford, Pa.

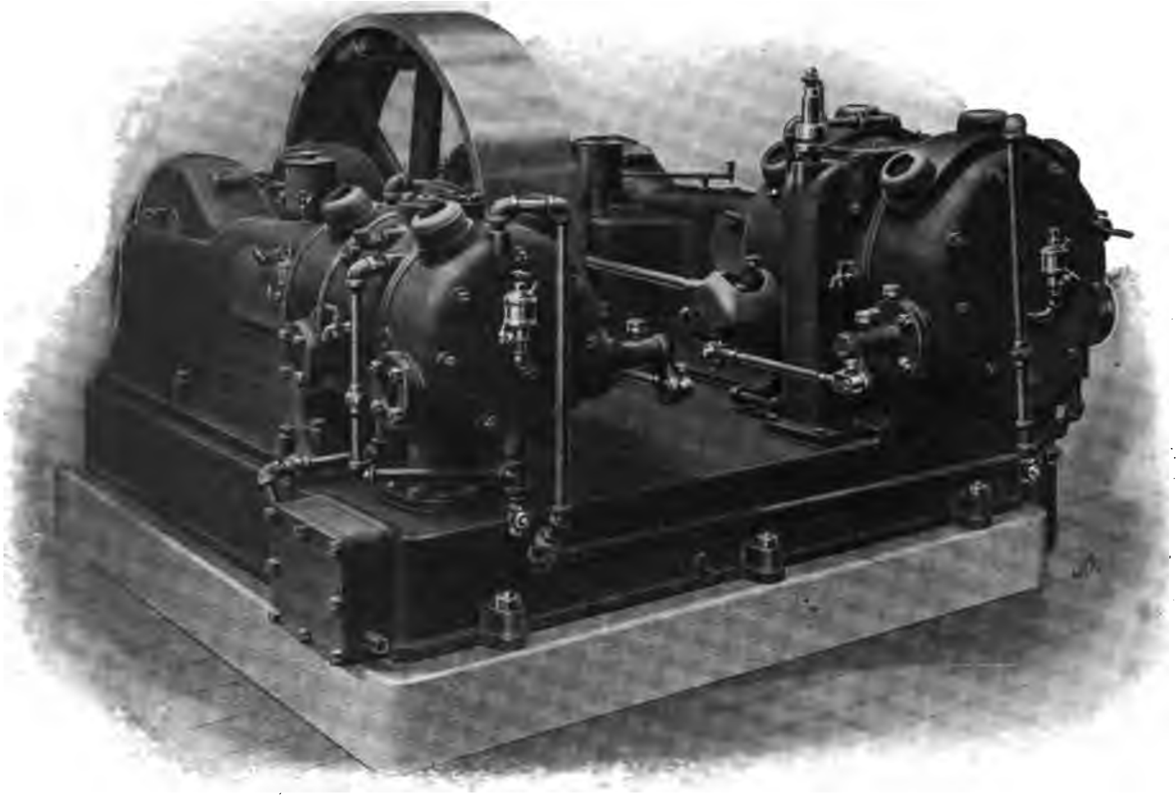


**Fig. 30. Class WJ Simple Duplex Belt-Driven Air-Compressor. Sullivan Machinery Co., Chicago. Ill.**

## POWER-DRIVEN COMPRESSORS

Power-driven compressors are an evolution from steam-driven types, and partake of the general characteristics of the latter. They differ, however, in one vital essential—namely, while steam-driven compressors are variable-speed, variable-load machines, power-driven compressors are in the vast majority of cases constant-speed variable-load machines. This arises from the fact that the latter are usually driven from some constant-speed prime mover; and it makes the problem of regulation quite distinct in the two cases. Before taking up this point in detail, however, the general subject of power-driven compressors will be examined.

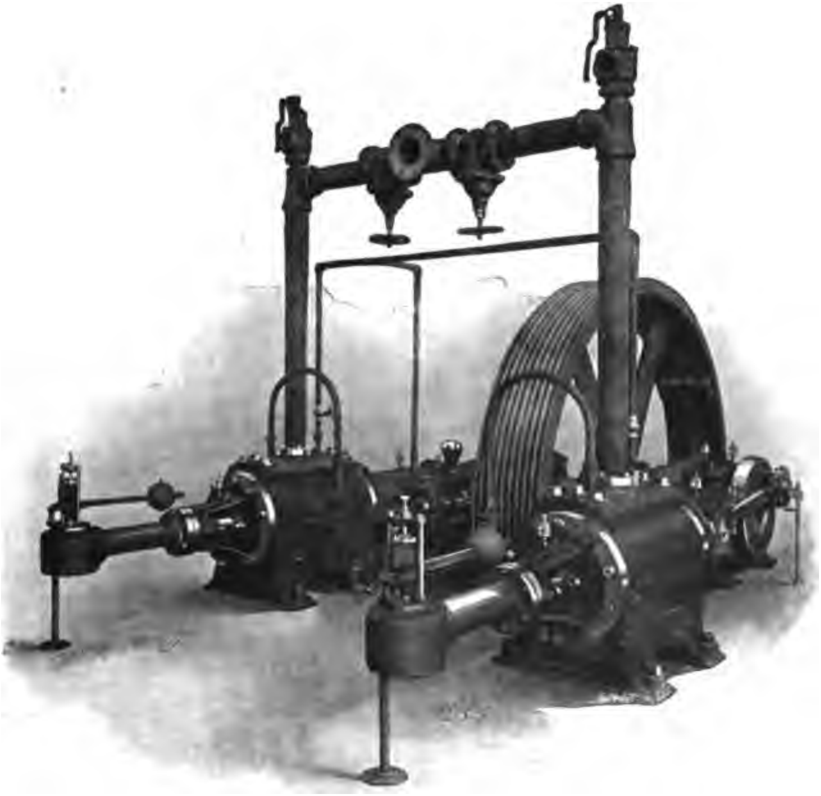


Fig. 31. Class D-1 Rope-Driven Simple Duplex Air-Compressor. Ingersoll-Rand Co., New York City.

*Belt drive* is always applicable except where a very short belt center, or distance between centers of driving and driven wheels, is necessary. In the latter case, so high a belt tension is required to avoid slipping of the belt, that an undue pressure on the shaft bearings is produced, resulting in excessive friction, low mechanical efficiency, and rapid wear of the bearing boxes. Belt centers should always be long enough to give the requisite belt contact with a comparatively slack belt; 20 to 25 feet is a good average, but 15 feet is perhaps the minimum which should be used. A good rule is to make the belt center three to four times the diameter of the larger pulley—preferably four times. The direction of belt motion should be from the top of the driving wheel to the top of the compressor wheel, this arrangement placing the slack belt on top and increasing the arc of contact.

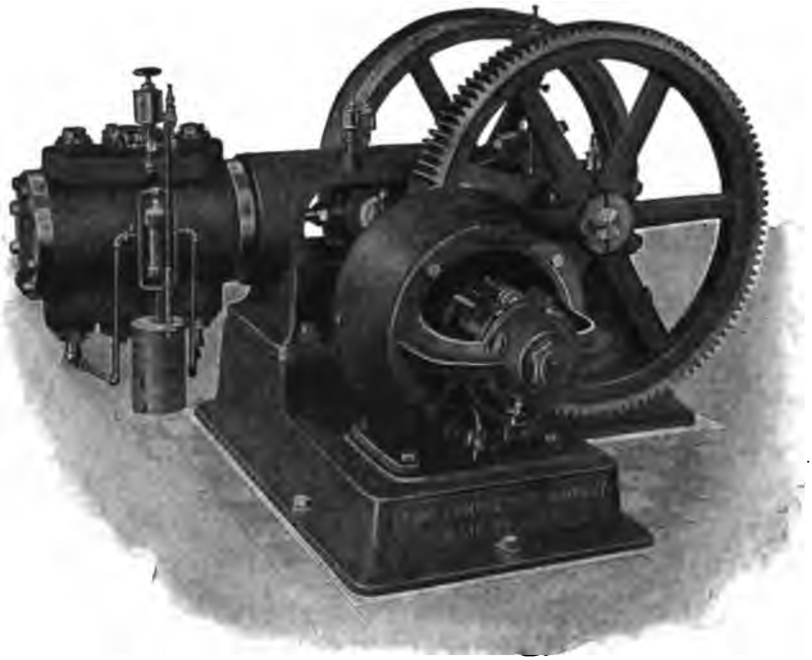


Fig. 32. Gear-Driven Straight-Line Single-Stage Compressor, Class BE.
Bury Compressor Company, Erie, Pa.

*Rope drive* is always applicable, though seldom used where the power to be transmitted is small. It is particularly advantageous where a short belt center only can be had, as the grip of the rope in properly shaped grooves on the wheels gives all the pulling power necessary without undue tension and without excessive bearing friction. The durability and efficiency of the rope drive depend upon a correct shape to the rope grooves, and upon the use of a suitable number of ropes so that excessive strain on any one rope will be avoided.

*Gear drive* and *silent chain drive* are never desirable, and should be used only where conditions imperatively demand them, as where a very compact compressor unit is necessary. They are ordinarily used on compressors driven by an electric motor; and as the efficiency of either method of drive depends very largely upon maintaining proper alignment, the motor in such cases is usually mounted rigidly on an extension of the compressor frame, or else a sole-plate should unite motor and compressor bases. The best practice places the limiting power for gear drive with a rawhide pinion, at 40 H.P.; for gear drive with a steel pinion, at 60 H.P.; and for silent chain drive, at 50 H.P.

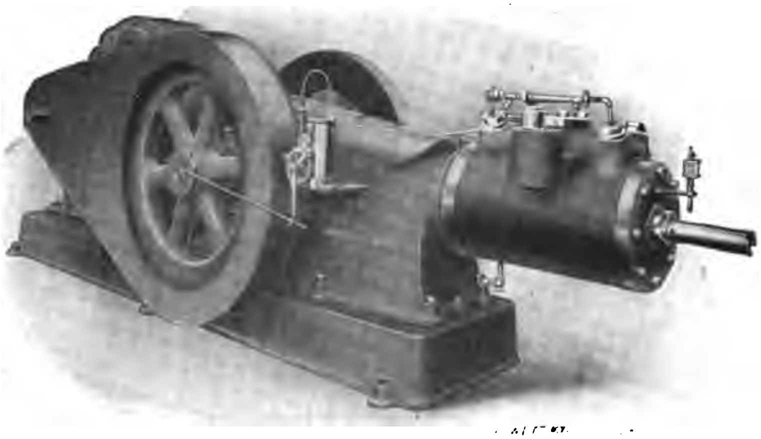


**Fig. 33. Class EE-1 Chain-Driven Single-Stage Electrical Air-Compressor. Ingersoll-Rand Co., New York City.**

The rules for proportioning belts, ropes, gears, and chains for compressor drive, are the same as those used in ordinary mechanical work.

A very modern development is the direct-connected electric-driven compressor, with the armature or rotor of a slow-speed motor mounted directly on the compressor shaft. This direct-shaft drive has also been used where the conditions of head and volume of a water power were such as to permit mounting an impulse wheel directly on the compressor shaft. These constructions are applicable only in units of comparatively large capacity, since the rotative speed is necessarily limited, and it is only in the large powers that motors can be used giving the required slow speed without excessive cost. The best machines of this class have very heavy mechanical structures, unusually large valve and port areas, and very effective cooling devices, so that a comparatively high speed (as compared with ordinary compressor practice) can be used without sacrifice of mechanical and volumetric efficiency. The direct-connected arrangement is the ideal one for electric or water-power drive, as the mechanical losses between motor and compressor are practically eliminated, the friction being little if any more than that of the compressor alone.

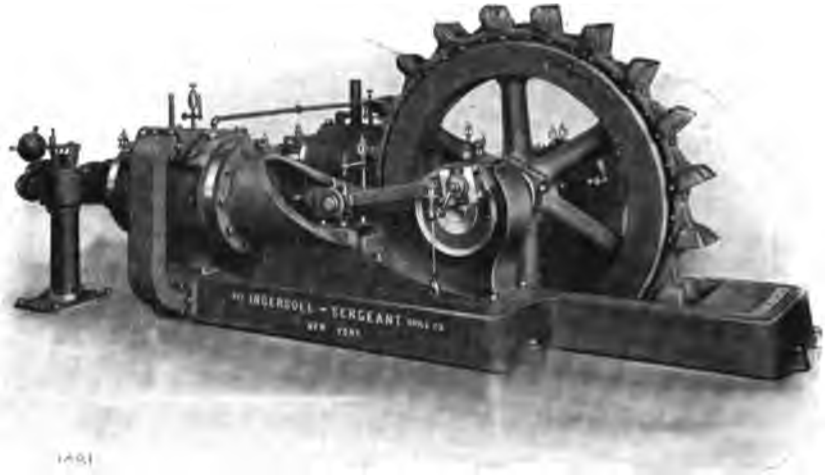


Fig. 84. Class J-2 Cross-Compound Air-Compressor Direct-Connected to Pelton Water Wheel.
Ingersoll-Rand Co., New York City.

**Regulation of Power-Driven Compressors.** Since the power-driven compressor is almost always a constant-speed machine, the methods of regulation and governing described for variable-speed

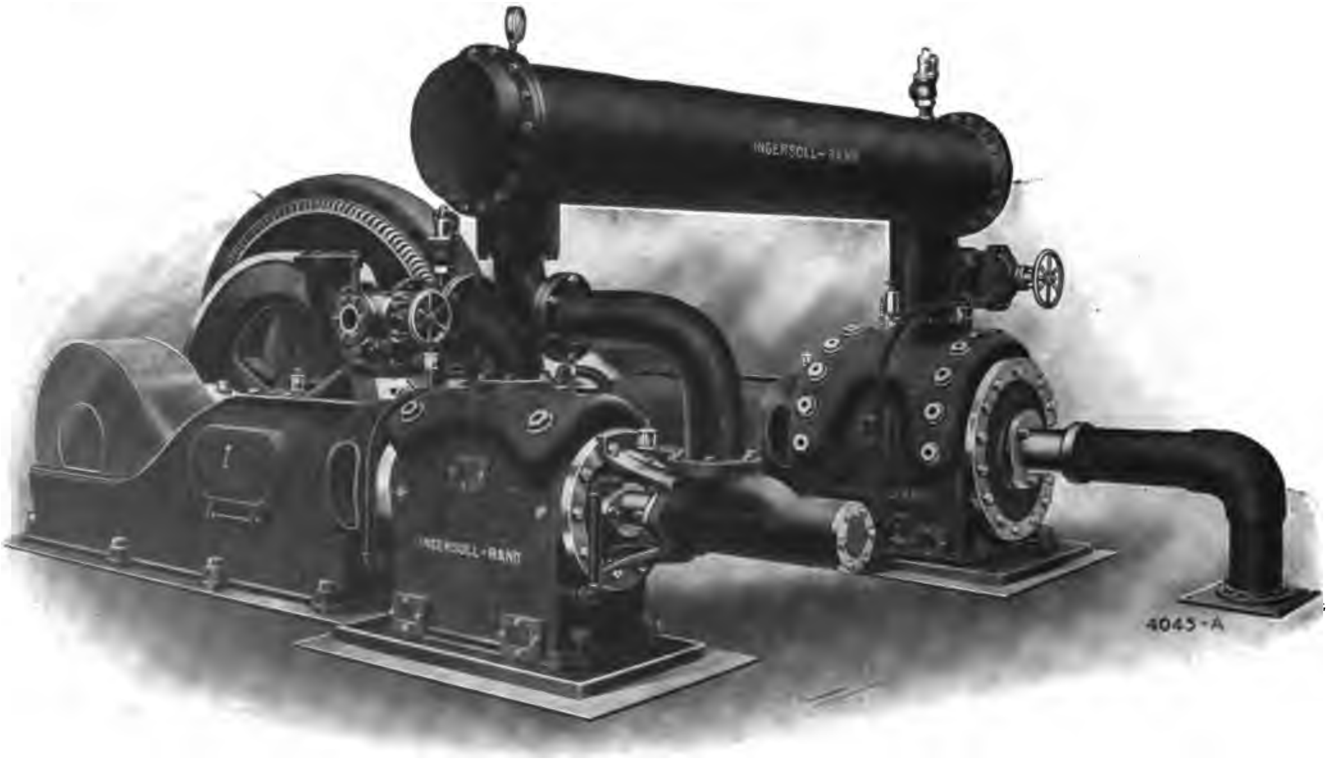


Fig. 35. Class PE-2 Direct-Connected Cross-Compound Electrical Air-Compressor. Ingersoll-Rand Co., New York City.

steam-driven machines evidently cannot here be applied. Constant speed means constant piston displacement; and the problem of delivering a variable volume of air with constant piston displacement, becomes one of making a portion of that displacement non-effective in the compression and delivery of air. Only the fundamental principles of several methods of accomplishing this will here be discussed.

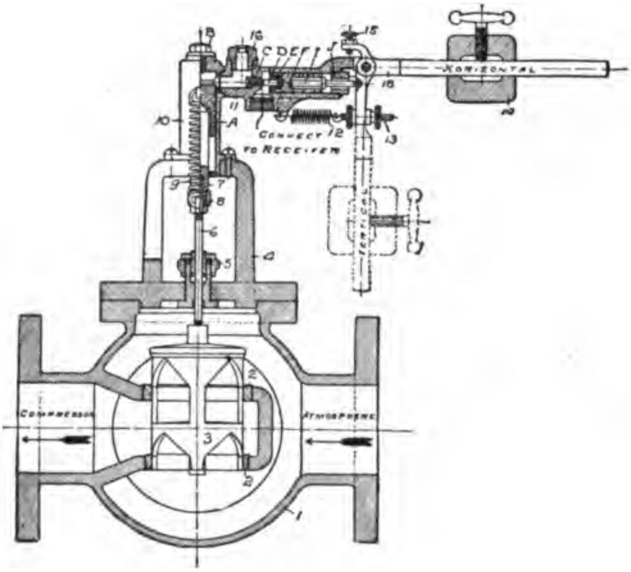


**Fig. 36.** "Erie" Unloader for Air-Compressor Work. Maintains a constant pressure in receiver by governing inlet of compressor to let in only exact amount of air used, and to load compressor in exact proportion to work required. Jarecki Manufacturing Company, Erie, Pa.

The first method is really one of *unloading*, rather than of regulating. A pressure-controlled mechanism is arranged so that when pressure exceeds normal, due to excess of delivered volume over demand, a communication is opened between the two sides of the compressing piston. Usually this is accomplished by opening and holding open one or several of the discharge valves at both ends of the cylinder, the air then simply sweeping back and forth from one side of the piston to the other through the open valves and the air-discharge passage. When normal pressure is restored, the valves are automatically closed, and compression and delivery are resumed. Evidently this is practically a total unloading of the machine for a longer or shorter period—a sudden release from load and a sudden resumption of load. Moreover, the air which is swept back and forth by the piston in its travel is air under full pressure; so that when the discharge valves suddenly close, the piston at once encounters a full cylinder of air at maximum pressure. These facts limit regulators of this class to machines of comparatively small capacity.

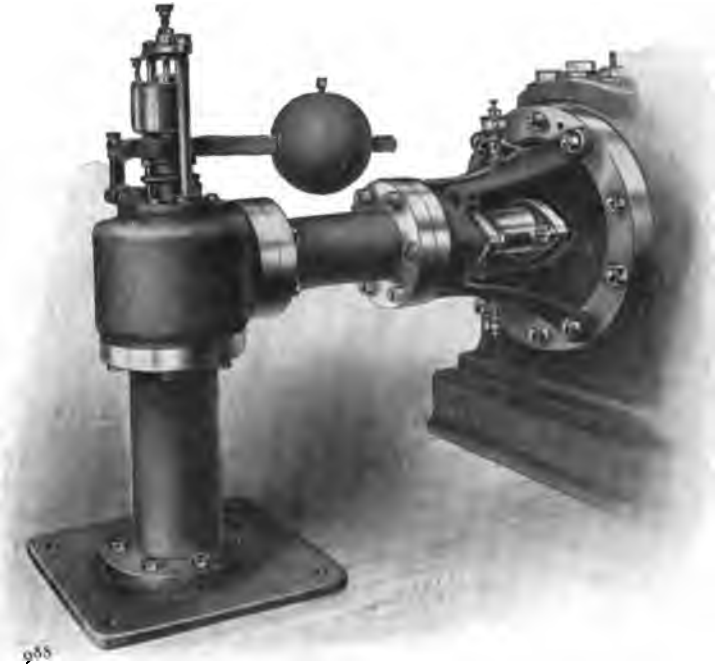
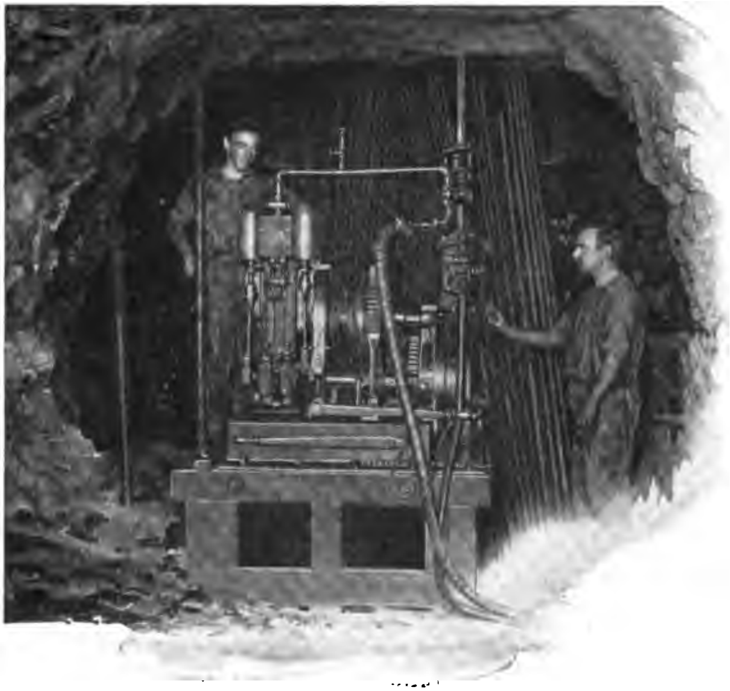


Fig. 37. Class A-18 Choking Controller for Power-Driven Compressors. Automatically proportions power to load by throttling the air intake as pressure rises above the fixed normal.
Ingersoll-Rand Co., New York City.

Another method provides, by means of a pressure-operated device, for the partial or total closing of the compressor intake under reduced load. To avoid the dangers attendant upon such an operation acting suddenly, these devices are provided with some damping mechanism so that they are compelled to operate slowly, making the release or resumption of the load gradual. The cutting down of the air intake results in a rarification of the air entering the



SULLIVAN COMPRESSED AIR DIAMOND DRILL

cylinder, and a greater range between initial and discharge pressures, with a corresponding increase in the range of temperatures. This method of regulation, therefore, is not suitable for very great load variations; nor is it recommended for such conditions by the builders responsible for it.

The third method is very similar to the first, except that here the inlet valves, instead of the discharge valves, are held open when the machine is unloaded, the piston thus simply drawing in and forcing out air at atmospheric pressure. It is open to the same criticism (though in somewhat less degree) as the first method—namely, undue shock and strain on release and resumption of load.

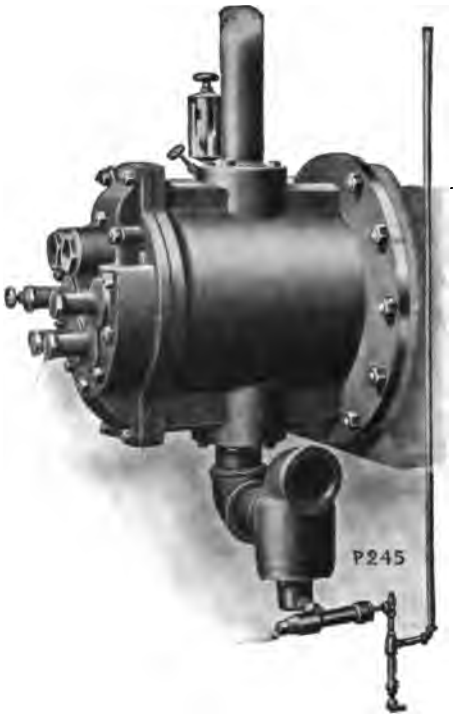


Fig. 38. "Imperial" Unloader for Steam- and Power-Driven Compressors. Ingersoll-Rand Co., New York City.

The fourth method uses a pressure-controlled valve on the compressor discharge of single-stage machines, combining also the functions of a check-valve to limit the escape of air from the receiver or air line. Excess pressure blows the discharge to atmosphere, instead of into the line. This arrangement is also used on two-stage machines by placing it on the low-pressure discharge to the intercooler. Then, when the governor valve is opened by excess pressure, the low-pressure cylinder discharges to atmosphere, and the high-pressure cylinder acts simply as a low-pressure cylinder with intake at atmospheric pressure. This device is more of a relief-valve than an unloader, for the piston must continue to compress to a pressure which will open the discharge valves; and this volume of compressed air, with its power equivalent, is wasted.

On power-driven compressors with Corliss intake valves, several different methods of unloading or regulating are used.

By one method, the Corliss valve is held open for the full admission stroke, and also for a part of the compression stroke, this latter portion being determined by the degree of unloading called for. Evidently this is practically equivalent to a shortening of the stroke of the compressor.

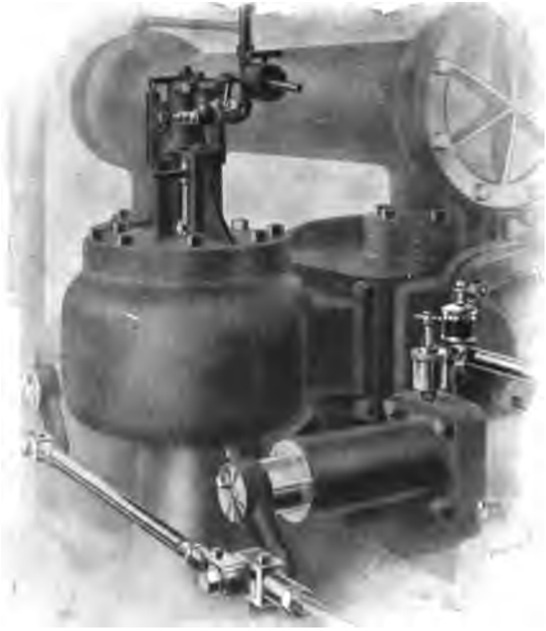


Fig. 39. Unloading Device for Steam and Power-Driven Air-Compressors. Sullivan Machinery Co., Chicago, Ill.

By another method the Corliss intake valve is opened full at beginning of admission, but closes later in the admission stroke. The air admitted to that point is expanded or rarified for the remainder of the compression stroke, and then compressed, the volume of compressed air delivered being of course reduced. This arrangement is productive of an excessive temperature range in the cylinder.

Still a third method opens and holds open the intake valves at one end of the cylinder, or at opposite ends in duplex machines.

The effect of this is to make ineffective one out of every two strokes. If still further unloading is necessary, the intake valves at the other end of the cylinder or cylinders are opened and held open.

The three arrangements just outlined all operate by a pressure-controlled mechanism which actuates some form of trip or release on the Corliss air-valve gear, somewhat similar to the release mechanism of the Corliss steam valve for varying the cut-off.

Three things are to be avoided in the successful unloader or regulator for power-driven compressors: *first,* a sudden release or resumption of load, throwing heavy strains on the machine; *second,* undue rarification of the intake air, resulting in a wide range of cylinder pressures and temperatures; *third,* the blowing-off of compressed air to the atmosphere with a waste of power.

In the regulation of the power-driven compressor, less reliance must be placed upon the automatic regulation of the individual machine than upon the intelligent subdivision of the load between two or more machines and the careful management of the resulting plant. In designing a plant of these machines, maximum capacity must be cared for in the normal full output of the machines, while partial loads are provided for by starting or stopping one or more machines, the remainder running at or very near full load. This question will be taken up later in the discussion of air-power plant design.

**Starting Unloaders for Power-Driven Machines.** It is usually desirable to start a power-driven compressor with no load, throwing on the load gradually after normal speed has been reached. This is in fact essential in machines driven by electric motors, for the heavy inrush of current in starting under load is dangerous, particularly where power is taken from a transmission circuit supplying other motors.

Evidently almost any of the unloading devices noted in the previous section can be used for this purpose if properly arranged for manipulation. The usual form, however, is simply a by-pass valve to atmosphere on the line close to the compressor, protected by a check-valve between it and the receiver to prevent the return of air from the line when the starting unloader valve is open. This check-valve is essential where several compressors serve one line, permitting cutting in or out any machine without unloading the others. This by-pass valve is opened on starting, when the compressor simply compresses to a pressure sufficient to open its discharge valves, this air escaping then to atmosphere. When normal speed is reached, the by-pass or unloading valve is gradually closed and load resumed. On two-stage machines, an unloader valve should be provided on the low-pressure discharge to the intercooler, as well as on the high-pressure discharge to the line. In the latter case, both cylinders operate momentarily as low-pressure cylinders.

## VERTICAL AND HORIZONTAL COMPRESSORS

The vertical air-compressor has never attained to any degree of popularity, except for comparatively small capacities and for special work where the smallest possible floor space is demanded, as in the high-pressure machines for torpedo service on warships. About the only advantage the vertical type has, is in its compactness; and that has never been as important a feature in compressor as in engine practice. When large capacities are reached, the horizontal type offers a decided advantage in its ready accessibility, whereas vertical compressors of equivalent capacity would tower above the floor-line out of reach for ready inspection and close attention. Vertical compressors are almost exclusively limited to the power-driven class; and in the smaller sizes, there are excellent machines of this type on the market. The horizontal type vastly predominates, however, even among small-sized units.

## STRAIGHT-LINE AIR-COMPRESSORS

The straight-line air-compressor is probably the original type among machines for this purpose; for, in the beginning of pneumatic development, the first logical step would have been to attach an air cylinder to a steam engine, with its piston on an extension of the steam piston-rod. In essentials this is all that a straight-line compressor is—one or more steam cylinders and one or more air cylinders arranged in a *straight line,* with their pistons on one continuous piston-rod. An example of this type in its simplest form, is the *air-pump* on a locomotive, compressing air for the air-brakes. Add a pair of fly-wheels to equalize operations throughout the stroke, and a typical straight-line compressor is the result.

The distinguishing characteristic of a straight-line compressor is the direct application of power to load, with a minimum intermediate loss. The best machines of this type are marked by the greatest simplicity of construction, and by a freedom from auxiliary and accessory mechanisms which absorb power in friction, require frequent attention, and offer possibilities of derangement and breakdown. In fact, the simplicity of the straight-line type is its chief claim for consideration; and when attempts are made to add complicated and questionable refinements to it, the excellence of the original type is likely to be sacrificed, with no material gain.

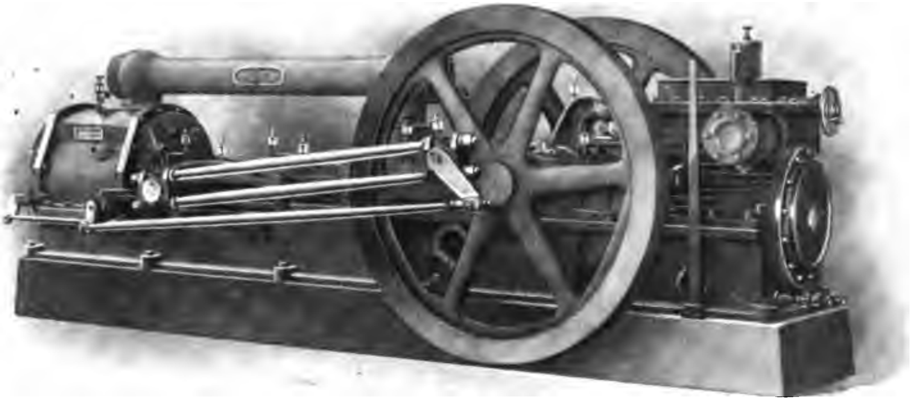


**Fig. 40. Straight-Line Steam-Driven Tandem Compound Air-Compressor. Norwalk Iron Works Company, South Norwalk, Conn.**

The compactness of the straight-line compressor, and the fact that it is usually a self-contained unit entirely supported by its bedplate or frame, are other strong arguments in its favor, adapting it for use with a very simple foundation (or no foundation at all, in emergency), making its proper installation the simplest matter, and affording the greatest ease of transportation as a complete unit. This probably explains the fact that the straight-line type has found its greatest application for mining and contract work, where the location is difficult of access and where the plant is more or less temporary in character.

There is no more sturdy or dependable compressor type than the best modern straight-line within its proper sphere, if the fundamental principles which distinguish it are clearly understood and closely adhered to. It is pre-eminently the machine for everyday service in small or moderate capacities under conditions where moderate first cost, ease of installation, reliability in the hands of the average engineer, and a good day-after-day economy are the main objects sought. For other conditions—particularly where the lowest operating economy is important—preference should be given to the duplex type, which will be discussed later.

There are some points about the straight-line compressor which are not beyond criticism; but the well-known advantages of th type usually outweigh these disadvantages in the class of service for which this machine is fundamentally intended. A plain, simple steam, single-stage, straight-line compressor is almost above reproach in its proper place. The addition of another air cylinder for two-stage compression, means a longer piston-rod, with more expansion and contraction under changes of temperature to be provided for in the cylinder clearance. If a second steam cylinder for steam compounding is added, this trouble is still further magnified. In the latter case the very long connecting rods required, also offer possibilities of trouble. But the simple steam tandem compound and the double tandem compound compressors are not to be altogether condemned, even on these grounds. There are first-class machines built in both these types.

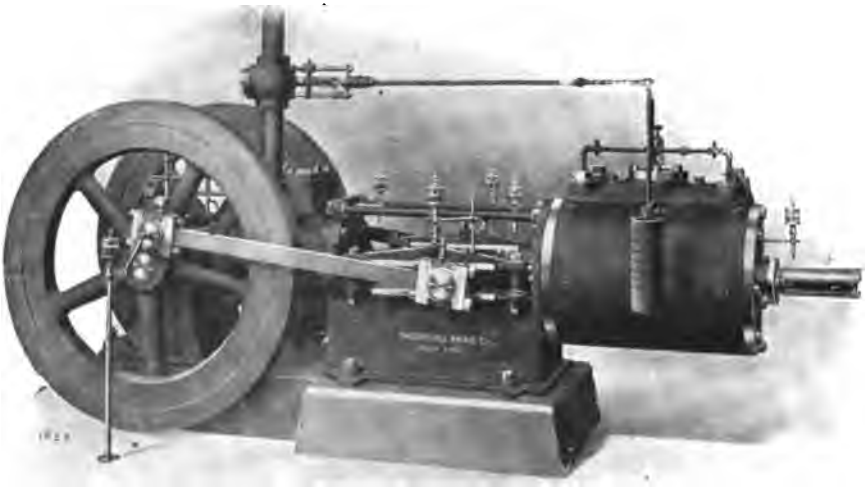


Fig. 41. Class F-1 Steam-Driven Straight-Line Air-Compressor. Ingersoll-Rand Co., New York City.

The double tandem compound straight-line cannot be considered wholly a success under the average conditions for which a straight-line machine would probably be selected. The inherent characteristic of the type—the direct opposition of power to resistance—produces conditions offering serious objections to this construction

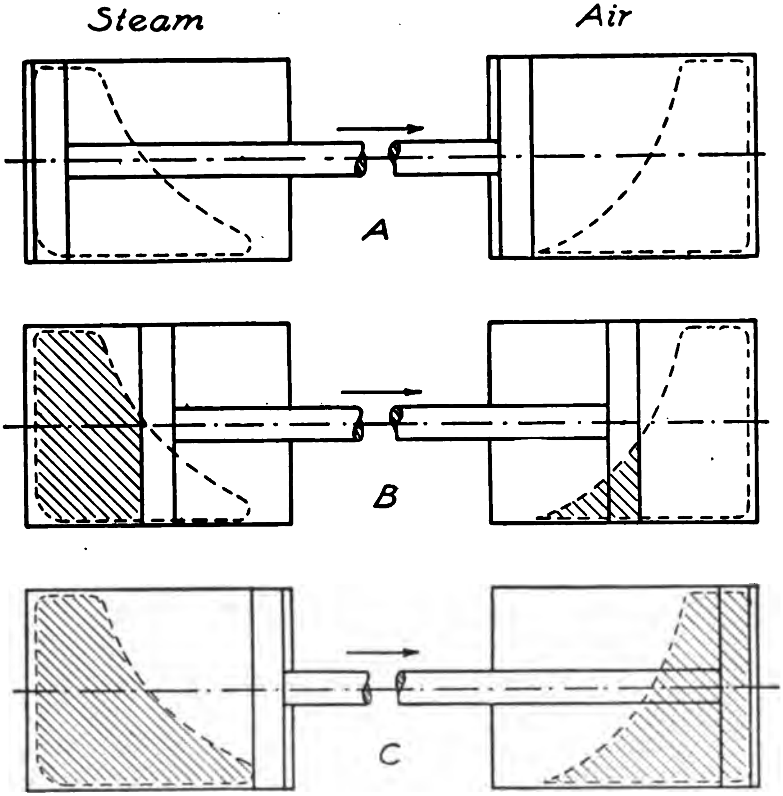


**Fig. 42. Pressure Relations in the Simple Steam Single-Stage Straight-Line Air-Compressor.**

under widely variable loads. This will be more clearly understood after an investigation of the pressure relations in the steam and air cylinders of the straight-line air-compressor.

*Pressure Relations in the Straight-Line Type.* Fig. 42 indicates diagrammatically the principle of the straight-line compressor with simple steam and air cylinders. The indicator cards of the two ends are only roughly drawn in, with no attempt at accuracy. In *A*, the

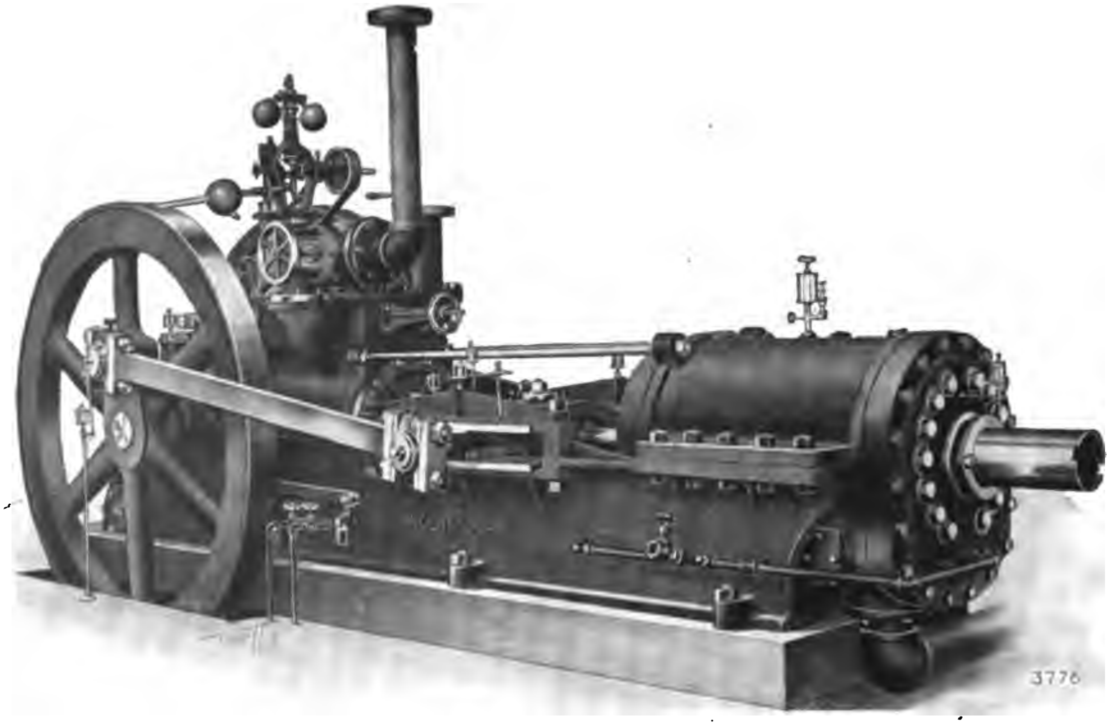


Fig. 43. Class A-1 Simple Steam Single-Stage Straight-Line Air-Compressor. Ingersoll-Rand Co., New York City.

stroke is just beginning, and boiler pressure is acting on the steam piston with maximum power, which will be maintained until the point of cut-off; but in the air cylinder the resistance due to compression is nothing at the beginning of the stroke; and even when cut-off has been reached in the steam end, the load on the air piston will still be very small. In *B*, the conditions at mid-stroke are shown, and steam pressure still exceeds air pressure. In *C*, the conditions at the completion of the stroke are shown; and from the moment the discharge point in the air cylinder was passed, a rapidly diminishing steam pressure has been applied to maximum air pressure.

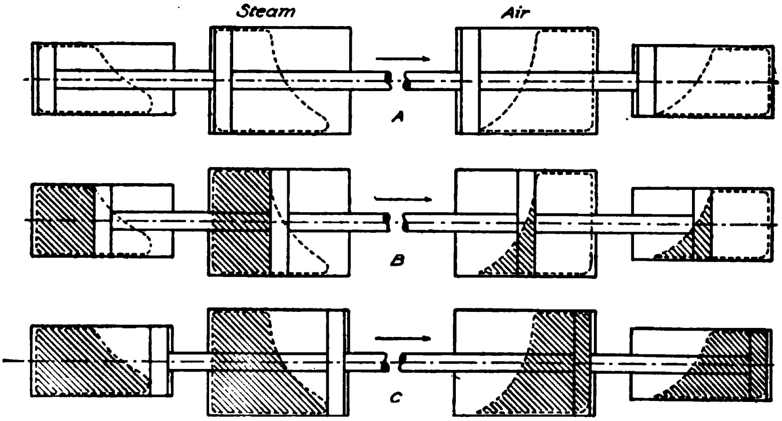


**Fig. 44. Pressure Relations in the Double Tandem Compound Straight-Line Compressor.**

Evidently there was surplus power during the first half of the stroke, and excess resistance during the last half. The condition is only rendered worse as cut-off in the steam cylinder is shortened to give better steam economy. Some method of equalizing the distribution of power and load must be used; and this is the function of the fly-wheels. They absorb the surplus energy during the first half-stroke, and deliver it against excess resistance during the last half-stroke; but their capacity for absorbing and giving out work depends upon their speed of rotation; and at slow speeds and short cut-off, even with very heavy fly-wheels, the straight-line compressor is apt to come to a dead stop. To be capable of regulation under wide load variation, therefore, the cut-off in the steam cylinder of the simple steam, single-stage air, straight-line

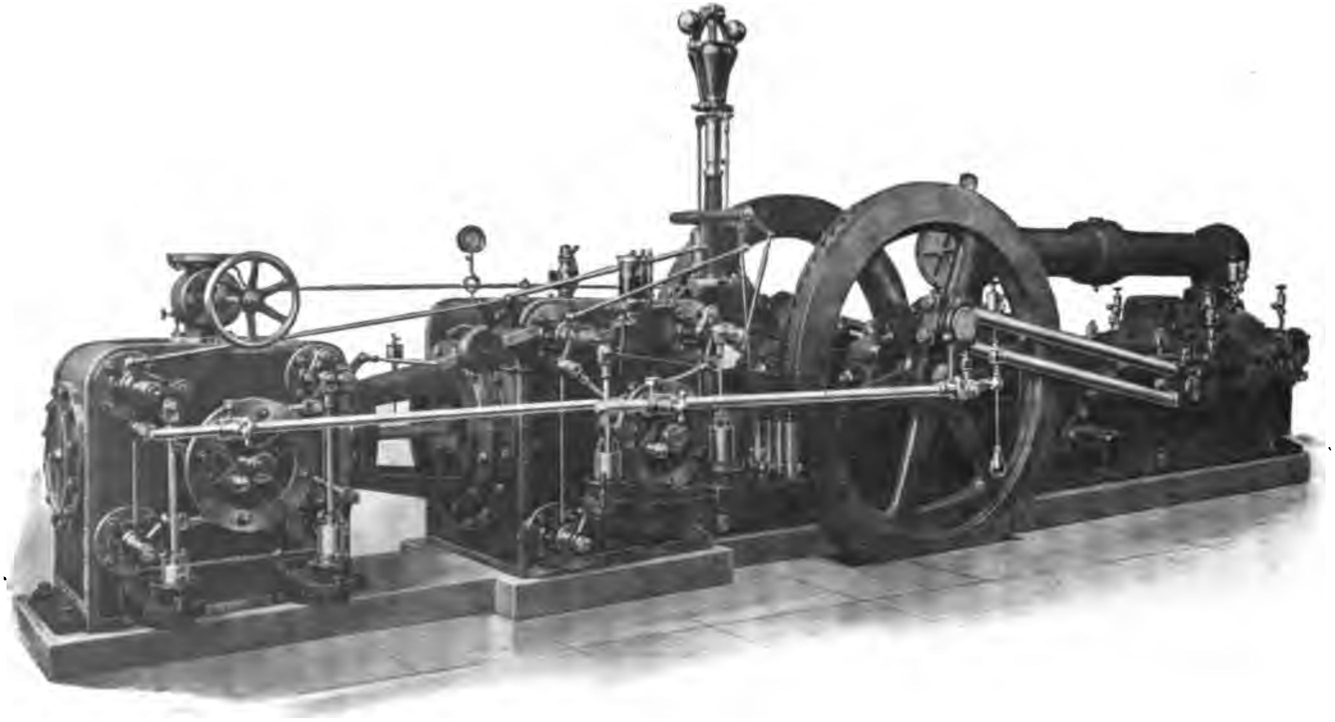


Fig. 45. Class WC Corliss Double Tandem Compound Air-Compressor. Sullivan Machinery Co., Chicago, Ill.

compressor must be late enough to permit the machine to run at slow speed, in which case there is a waste of steam at more nearly full load and at full load.

The pressure relations in the double tandem compound are sketched in Fig. 44, along the same lines as in the previous figure. The secret of compound steam economy lies in properly proportioned cylinders and in the correct adjustment of cut-off to give exactly the right expansion in both cylinders. If the cut-off is changed under varying speed to meet the conditions already described, it is evident that the ratio of expansions is deranged and the economy accordingly sacrificed. These facts practically forbid the successful compounding of the steam end of a straight-line compressor, except for operation under fairly steady loads. Even the plain simple steam single-stage straight-line is not economically self-regulating at less than 40 or 45 per cent of full load. Nevertheless the straight-line compressor, even in its more refined types, remains among the most popular machines for all-around service under fairly constant load, or in conditions where the mechanical superiorities of the type are of more importance than the very highest steam economy.

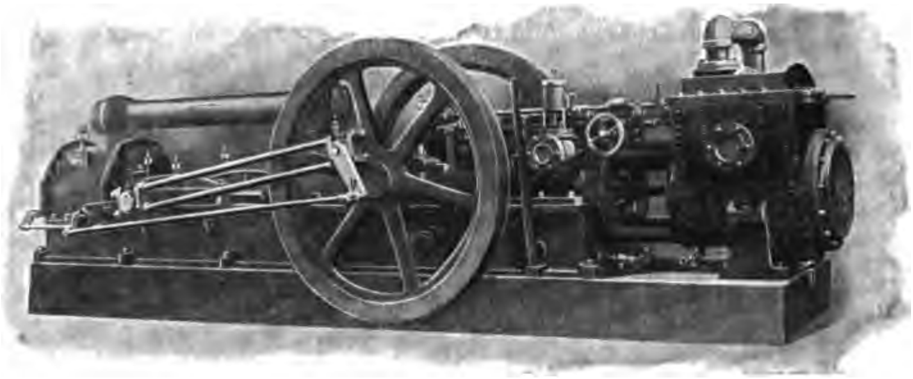


Fig. 46. Straight-Line Double Tandem Compound Air-Compressor.
Norwalk Iron Works Company, South Norwalk, Conn.

### DUPLEX AIR-COMPRESSOR

The duplex compressor may be best described as two straight-line machines side by side, with a common shaft for the two and a fly-wheel mounted between them. With this understanding,

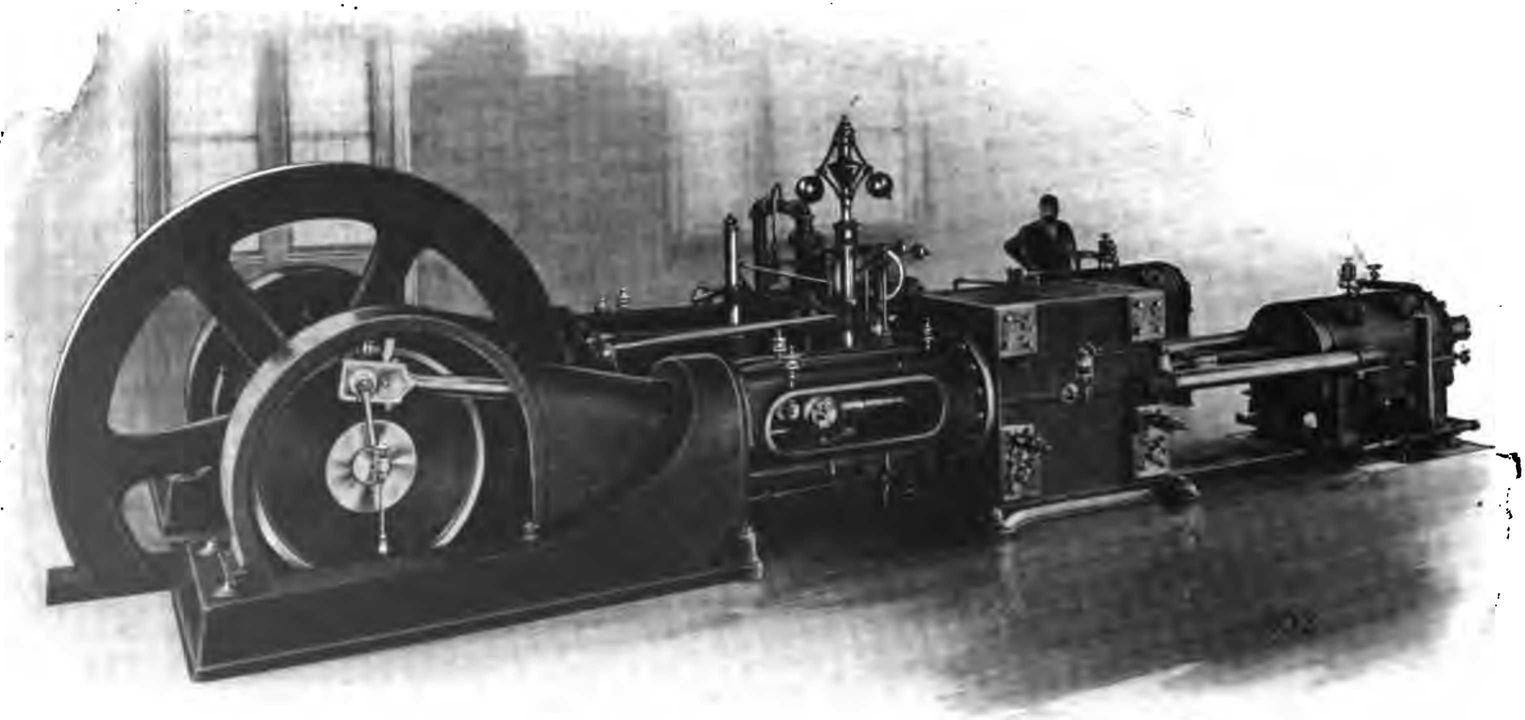


Fig. 47. Class C-4 Corliss Cross-Compound Duplex Air-Compressor.
Ingersoll-Rand Co., New York City

it may be stated at the outset that the duplex compressor has practically all the good features of the straight-line, with some additional advantages distinctly its own. Of course the compactness which characterizes the straight-line machine is sacrificed in the duplex compressor, as well as the portable features. Usually there are one steam cylinder and one air cylinder in each half of a duplex, either or both being cross-compounded. For three- or four-stage compression, there may be two air cylinders on each side; and sometimes duplex two-stage compressors are built with a high- and low-pressure air cylinder on each side arranged in tandem, with each side delivering one-half the total capacity in two-stage compression.

The duplex compressor is built with quartered cranks; that is, the crank-pin of one side is set one-quarter of a circle, or 90 degrees, in advance of the other crank-pin. The reason for this will be obvious upon examination of the four diagrams, Figs. 49 to 52, which show the pressure relations in a duplex compressor at four equal points in a revolution. For the sake of clearness, pressures on only one side of the pistons are here considered, for the back-pressure on the steam pistons, and the suction or intake pressure on the air pistons, do not materially affect the relations of power and resistance as here shown.

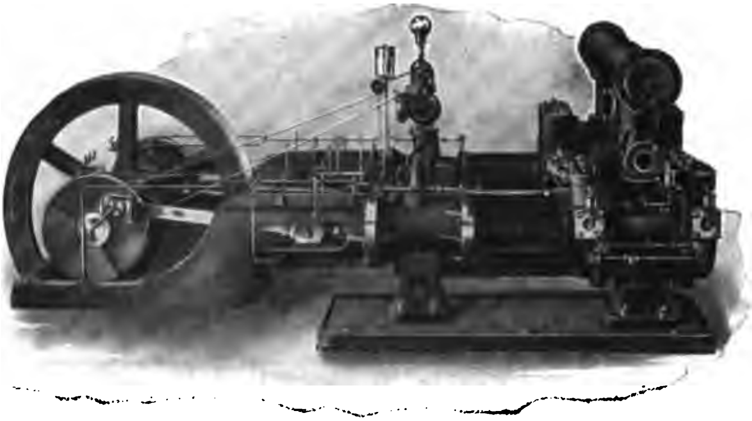


Fig. 48. "Cincinnati" Duplex Steam Cross-Compound Air-Compressor.
Laidlaw-Dunn-Gordon Company, Cincinnati, Ohio.

*Pressure Relations in the Duplex Compressor.* In Fig. 49, the pistons in the two lower cylinders are at the beginning of their stroke, and those in the upper cylinders are at mid-stroke. In the lower cylinders, there is no power and no load; in the upper cylinders the power greatly exceeds load, as indicated by shaded areas in the diagrams. Even if there were no fly-wheel, there would still be power enough to spare in the upper section to carry the lower section of the machine past its dead center, until steam was admitted to the lower steam cylinder.

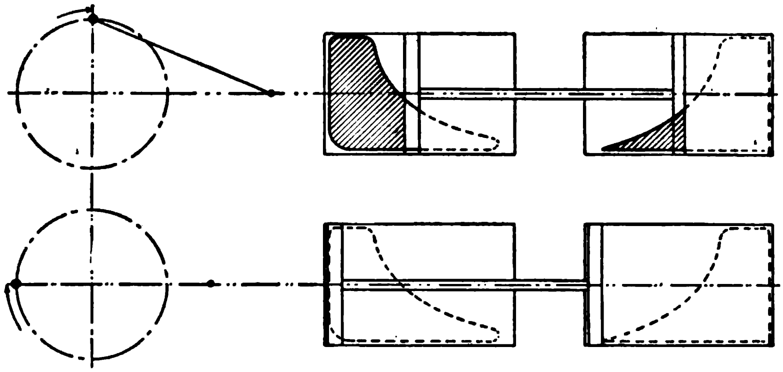


Fig. 49. Pressure Relations in Duplex Compressor, First Quarter.

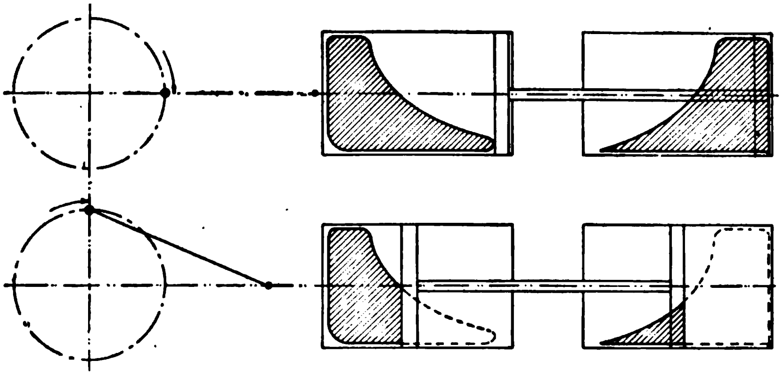


Fig. 50. Pressure Relations in Duplex Compressor, Second Quarter.

In Fig. 50, the cranks have advanced a quarter of a revolution; and in the upper section, just completing its stroke, power has fallen to the minimum, while load has risen to the maximum, the upper section being now on its dead center. In the lower section, conditions are identical with those shown in the upper section of Fig. 49, and surplus power is available to pull the upper section past its center.

In Fig. 51, after the completion of a half-revolution, the conditions are exactly reversed from those in Fig. 50; and in Fig. 52, after three-quarters of a revolution, the conditions shown in Fig. 50 are reproduced in inverse relation. On completion of the revolution, this cycle is exactly repeated.

It is evident from the foregoing, that there is no point in the compression cycle of a duplex compressor at which there is not ample power applied to the load. Such a machine would probably run, even without a fly-wheel; but the latter is always used to equalize the variations in the application of power to resistance, making the machine run smoothly and uniformly, whereas, without a fly-wheel, it would probably hesitate at or near dead centers. The fly-wheel, however, need not be so heavy in proportion for a duplex machine as for a straight-line compressor.

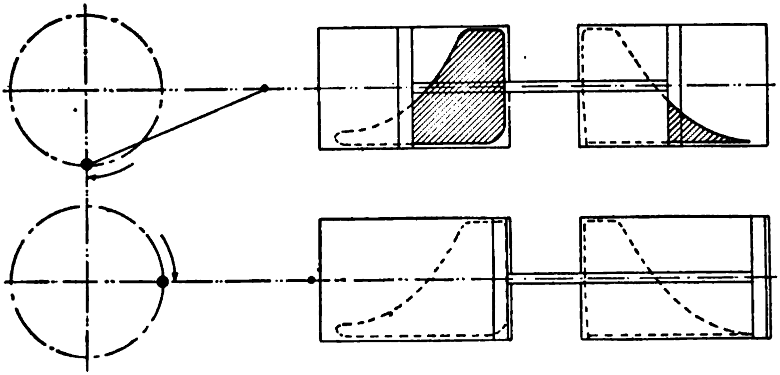


Fig. 51. Pressure Relations in Duplex Compressor, Third Quarter.

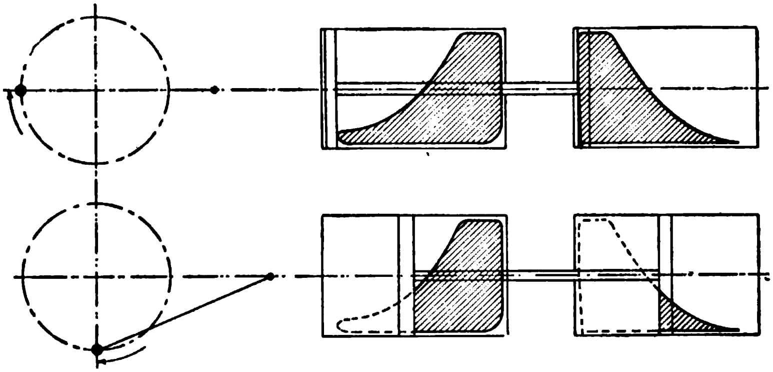


Fig. 52. Pressure Relations in Duplex Compressor, Fourth Quarter.

If the cut-off in the steam cylinders of a duplex compressor is not less than one-half, there is no time during the revolution when steam at boiler pressure is not being admitted on some face of the

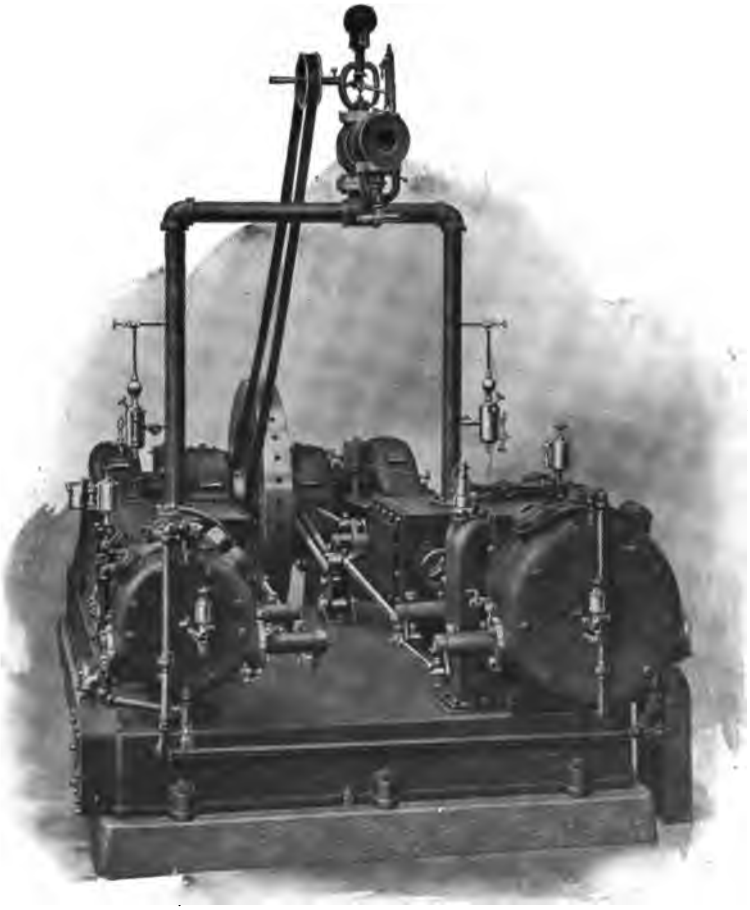


**Fig. 53. Class WE Duplex Steam Two-Stage Air-Compressor. Sullivan Machinery Co., Chicago, Ill.**

steam pistons; but so late a cut-off as this would probably not be used, except where the steam pressure was low. Even with a cut-off of three-eighths or one-quarter, the interval between steam admissions is so short that there is always ample power, taking into consideration the reserve held by the fly-wheel. It will be noted,

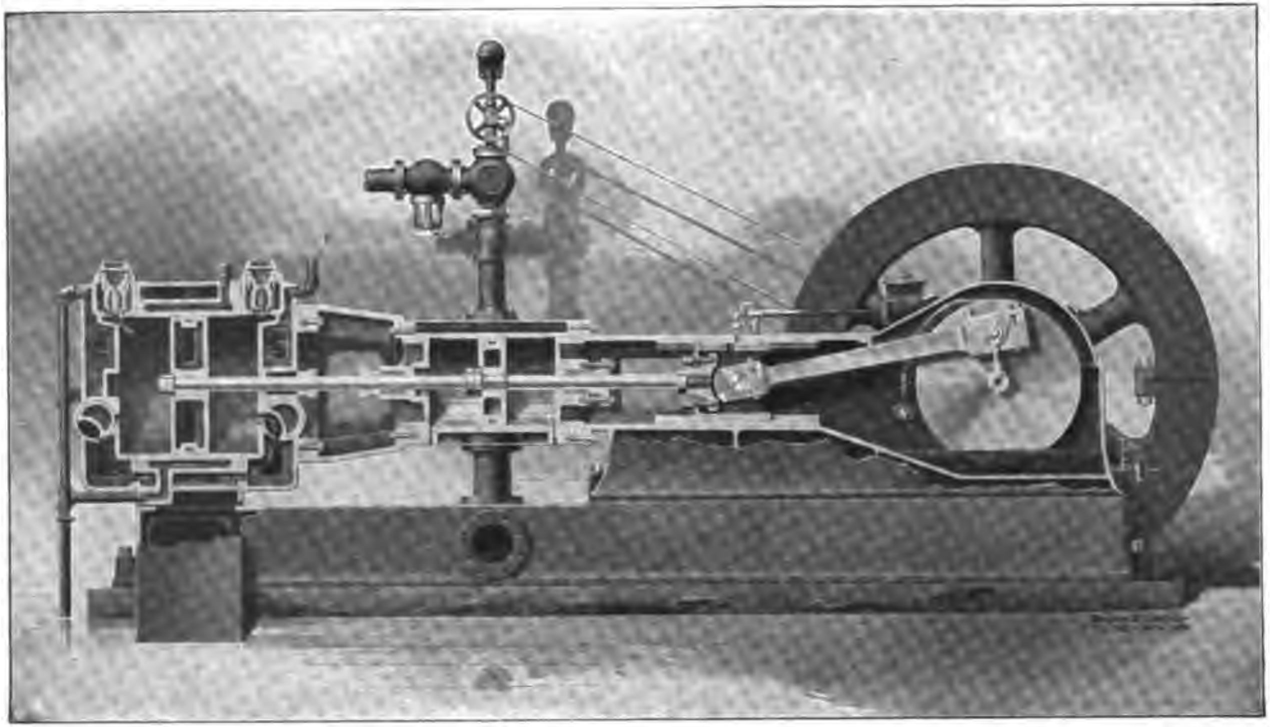


Fig. 54. Class WE Duplex Steam Two-Stage Air-Compressor, Sectional View Showing Oiling System.
Sullivan Machinery Co., Chicago, Ill.

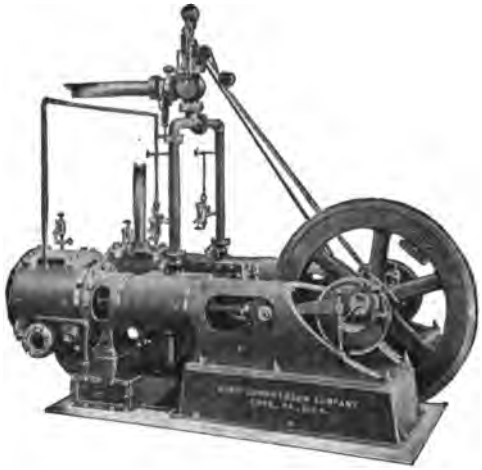


Fig. 55. Duplex Steam-Driven Compressor, with Duplex Steam and Cross-Compound Air Cylinders, Class CCS. Bury Compressor Company, Erie, Pa.

moreover, that this is true whether the speed is slow or fast, so that it is next to impossible to "stall" a duplex, even at the lightest load and it slowest speed. In practice, in starting a duplex machine with adjustable cut-off steam valves, the cut-off is usually lengthened until normal speed and full load are reached, after which the cut-off is reduced to the most economical point; and thereafter the machine will take care of itself at from a few revolutions per minute to full speed, responding instantly and automatically by means of its governor to any change in load.

**Advantages of Duplex Construction.** The first advantage of the duplex compressor lies in the fact that the quartered arrangement of cranks, with a suitable fly-wheel (which need not be so heavy as on a straight-line machine of equivalent capacity), equalizes the

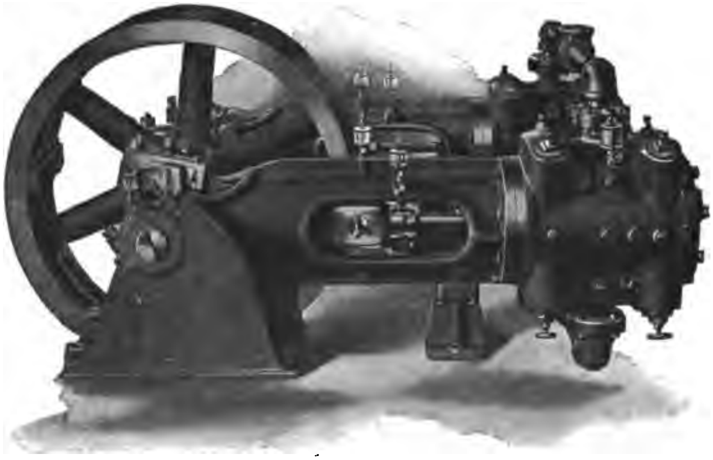


Fig. 56. Class SB Duplex Belt-Driven Air-Compressor. Laidlaw-Dunn-Gordon Co., Cincinnati, Ohio.

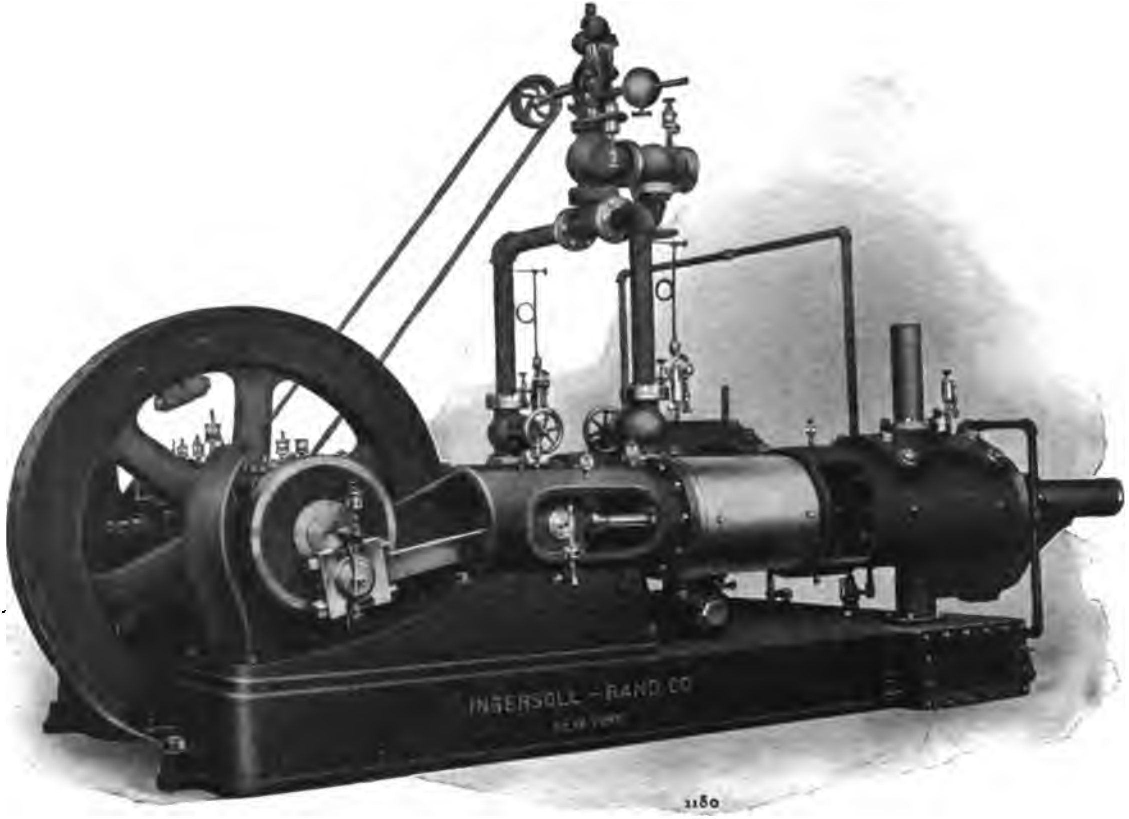


Fig. 57. Class H-2 Steam-Driven Duplex Air-Compressor, with Duplex Steam and Cross-Compound Air Cylinders. Ingersoll-Rand Co., New York City.

application of power to resistance, resulting in a more uniform distribution of the pressures in the cylinders, and in a steadier, more uniform operation.

The second advantage lies in the improved regulation resulting from the fact that there is always power available to be applied to the resistance at any point in the stroke and at any speed.

The third advantage of the duplex is that the mechanical efficiency of a well-designed machine of this class will be higher than that of a straight-line machine of equal capacity, owing to the equalization of stresses throughout the structure by the quartered arrangement of cranks. This advantage, however, presupposes that the machine has been properly lined up and leveled on its foundation, so that the main bearings run free, without undue friction resulting from incorrect alignment. A duplex compressor, unless it is of a pattern having a continuous bed-plate or sub-base under all parts, is more dependent upon its foundation than a straight-line machine.

The fourth advantage of the duplex type lies in the fact that in it all the advantages of compounding steam and air cylinders can be secured without the addition of more cylinders, valves, and wearing parts than are required in the simple duplex. It has been seen that a straight-line compressor cannot be compounded on either steam or air end without the addition of parts and without running into difficulties peculiar to the type. As a matter of fact, it is quite probable that a double cross-compound duplex will show a higher mechanical efficiency than a simple duplex or a double tandem compound straight-line; for the terminal or maximum stresses on bearings, etc., are reduced by compounding, resulting in a lower friction load. These reduced stresses have already been mentioned in connection with compound steam cylinders (page 44), and the point will be taken up later in its relation to compounded air cylinders.

The fifth advantage of the duplex construction is that full compound steam economy is secured over the entire load range, because the 90-degree crank arrangement assures operation whatever the load, without greatly departing from the point of cut-off of maximum steam economy. A double cross-compound air-compressor will practically maintain its best economy under all loads.

The sixth advantage of the duplex comes from its balanced con-

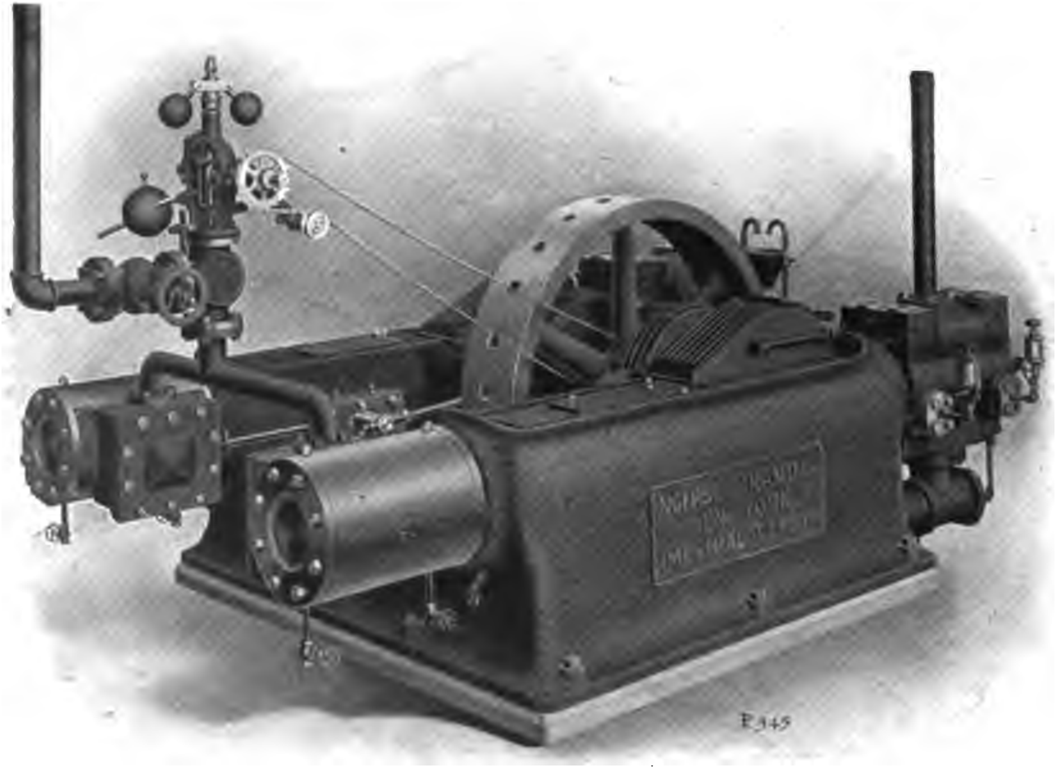


Fig. 58. "Imperial" Type X-2 Duplex Air-Compressor, with Duplex Steam and Cross-Compound Air Cylinders. Ingersoll-Rand Co., New York City.

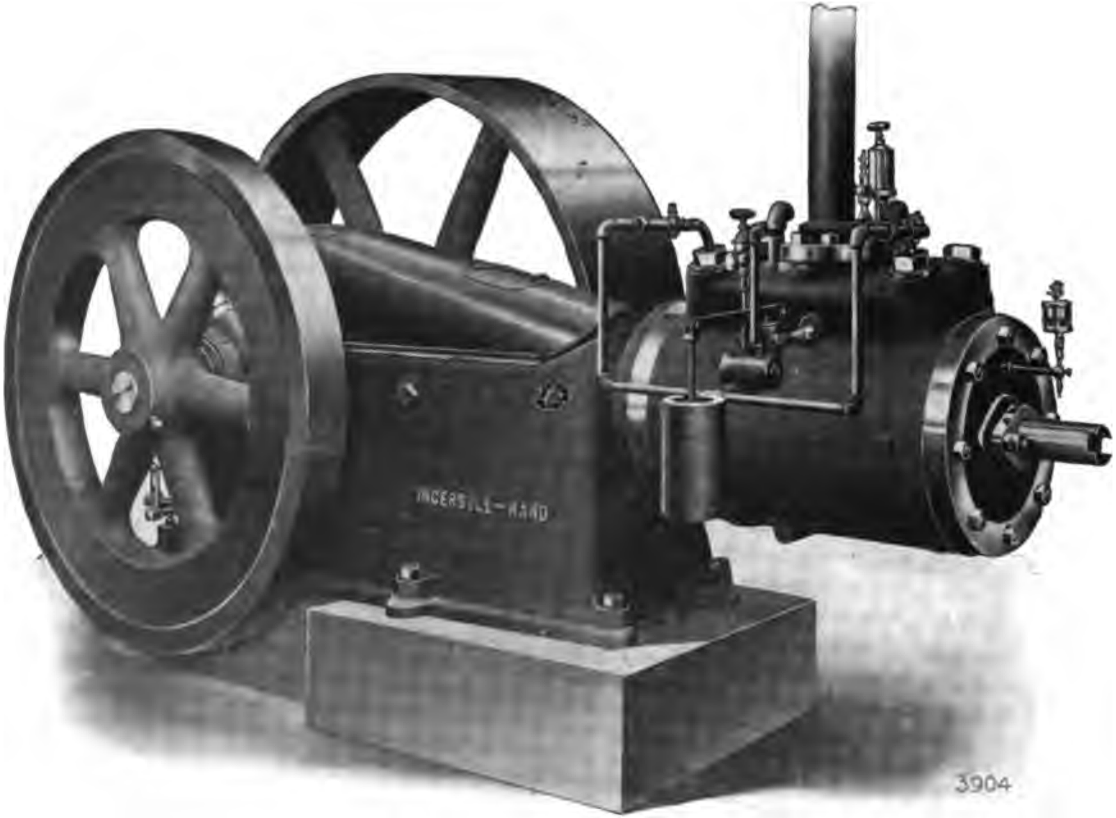


Fig. 59. Class EE-1 Straight-Line Single-Stage Belt-Driven Air-Compressor. Ingersoll-Rand Co., New York City.

struction, in which an impulse in one direction on one side is partially or wholly equalized by an impulse in the opposite direction on the other side. The combined effect of these opposing influences is a steady and minimized strain on the machine structure, reducing friction and tending toward a long period of service without breakdowns or interruptions and with the lowest repair cost.

The seventh advantage appears in the probability that an accident to a duplex machine will affect only one side; and in emergency the injured half may be disconnected and the other half run as a simple straight-line machine with reduced output, often tiding over a critical period. Occasionally, also, it is desirable to install only one side of a duplex compressor; and later the other half can be added—compounded if necessary—when a complete double cross-compound is had.

Only one disadvantage has ever been rightly urged against the duplex construction, and that is its dependence upon its foundation; but even this point is usually exaggerated, and certain modern duplex compressors have a continuous sub-base under the entire machine, making them as self-contained and as independent of their foundation as the simplest straight-line. While it is a fact that a duplex compressor occupies somewhat more floor space than a straight-line machine of equivalent capacity, yet the question of floor space has not as yet become of sufficient importance in compressor practice to make this point serious when compared with the many well-known advantages of the duplex type.

## WATER JACKETS

The necessity for some means of cooling the air during compression, has already been referred to (see page 8). Compressor practice of to-day uses one or both of two methods for securing this result—namely, (1) *water-jacketing;* (2) *compound* or *stage compression.*

In modern air-compressors, a space is provided around the barrel, and, if possible, around the heads of the air cylinders, through which water as cool as can be had is circulated in order to carry away the heat produced in compression, with the object of bringing the compression curve more nearly to the isothermal line. If the piston were made to move very slowly, so that this water could absorb and carry off all the heat, it is evident that this jacketing process would save all the work represented on the indicator diagram by the area between the actual compression line and the isothermal curve. In practice, however, a compressor cannot be run so slowly, and water-jacketing is only a partial solution of the problem of cooling.

While temperatures increase with the pressure in the cylinder, it is evident that the advancing piston steadily reduces the area of cool cylinder walls with which the air is in contact. Near the end of the stroke, when pressure and temperature are highest, only a very little of the walls is exposed, and the greater part of the cooling area remaining is in the cylinder-head. This emphasizes the importance of head jacketing, and the superiority of those constructions in which the air-valves are elsewhere than in the heads, thus affording the maximum percentage of cooled head area.

Moreover, air is a poor conductor of heat, so that even under the best conditions the heat from the interior of the air volume is not given up. On the contrary, only the external film in contact with the cylinder walls and head is cooled in any degree.

All things considered, therefore, water-jacketing an air cylinder realizes only a part of the saving of power which in theory it promises. Some beneficial effect in this respect, it certainly has—particularly in slow-speed, long-stroke machines; but the real problem of cooling during compression must be solved by other means, which will be discussed in the next section.

Cylinder jacketing is nevertheless essential in air or gas compression. Aside from the economical consideration just explained, it has other important and beneficial effects upon the satisfactory operation—and, indeed, upon the economy—of the compressor.

Not the least important of these is its effect upon the lubrication and therefore upon the wearing qualities of a machine. The compression of air from sea-level pressure and 60° F., to six atmospheres or 73.5 lbs. gauge, produces (unless some means of cooling is provided) a temperature of about 419° F. Such a temperature as this is destructive of all ordinary lubricants and packings. The former would be charred to a hard, coke-like substance, and the latter would be burned out. The inevitable result would be a rapid wear of piston rings, cylinder, rods, and valves, with extravagant leakage, loss of capacity, increased friction, and excessive power consumption. But the water jacket keeps all these parts reasonably cool, resulting in tight joints, free lubrication, little wear, reduced friction, easy running, and sustained capacity and efficiency.

An attempt to run continuously without jackets would heat the cylinder, piston, valves, rods, etc., to such a degree that dangerous expansion and contraction strains might be produced, resulting in distortion of parts, added friction, probable binding, possible breakage, and almost certain loss of capacity. The alternate expansion and contraction between the extreme temperatures of operation and shut-down, would make it practically impossible to maintain tight joints, and heavy leakage loss would certainly follow.

It has already been stated that it is almost as important that air should be admitted cool into the cylinder as that it should be kept cool during compression. If a water jacket is not used, the heating of the cylinder, piston, rods, valves, ports, head, etc., means that the air will be heated during admission, first, while entering through hot valves and ports; and second, by contact with a hot cylinder and piston inside the cylinder. This means a rarification of the entering air, reduced cylinder capacity, lower efficiency of compression, and excessive terminal temperature. Since jacketing is at best only partially effective, this latter argument suggests the advantage of those constructions which admit air through one, or at most only a few, large openings where only the outer film of air is heated, rather than through a number of small openings through which the air is finely subdivided and greatly increased in temperature.

The water-jacketing of compressor cylinders is thus seen to be essential on the grounds of power saving, large capacity, better wearing qualities, and a generally improved performance.

## COMPOUND AIR-COMPRESSION

If at several points in the stroke the compressing piston could be stopped, and the air already partially compressed and heated could be withdrawn long enough to be cooled by some external means to its initial temperature (or even lower), and then returned to the cylinder to be further compressed, heated, and again cooled, it is evident that a fairly uniform temperature could be maintained in this air volume over the full range of pressures, and the result would be very close to isothermal compression.

Evidently practical considerations forbid such a repeated starting and stopping of the piston; but the same results may be obtained by carrying on the process of compression in several cylinders, in the first of which moderate pressure and temperature are reached, and the air in this condition discharged through a cooling device which restores its initial temperature, then admitted to another cylinder where it is further compressed and heated, again withdrawn and cooled, again compressed—and so on until the desired terminal pressure is reached. Such a process, developed to a practical working cycle of operations, constitutes *compound* or *stage compression*, which is almost universally employed to-day when pressures are fairly high.

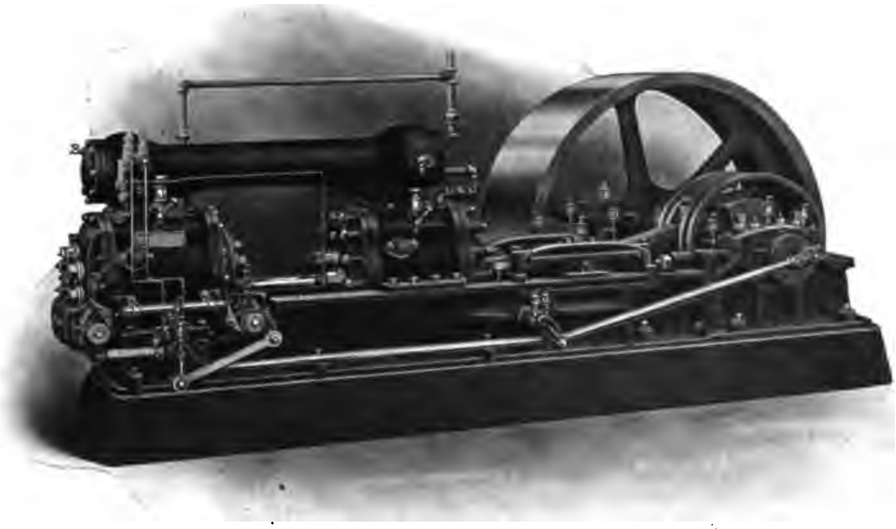


Fig. 60. Straight-Line Belt-Driven Two-Stage Air-Compressor.
Norwalk Iron Works Company, South Norwalk, Conn.

While the limiting pressures for which various degrees of stage compression are advisable, are not definitely fixed, an average of the practice of the leading builders seems to give the following:

For pressures up to 70 lbs., and under some conditions to 100 lbs., single-stage compression.

For pressures of 75 to 500 lbs., compound or two-stage compression.

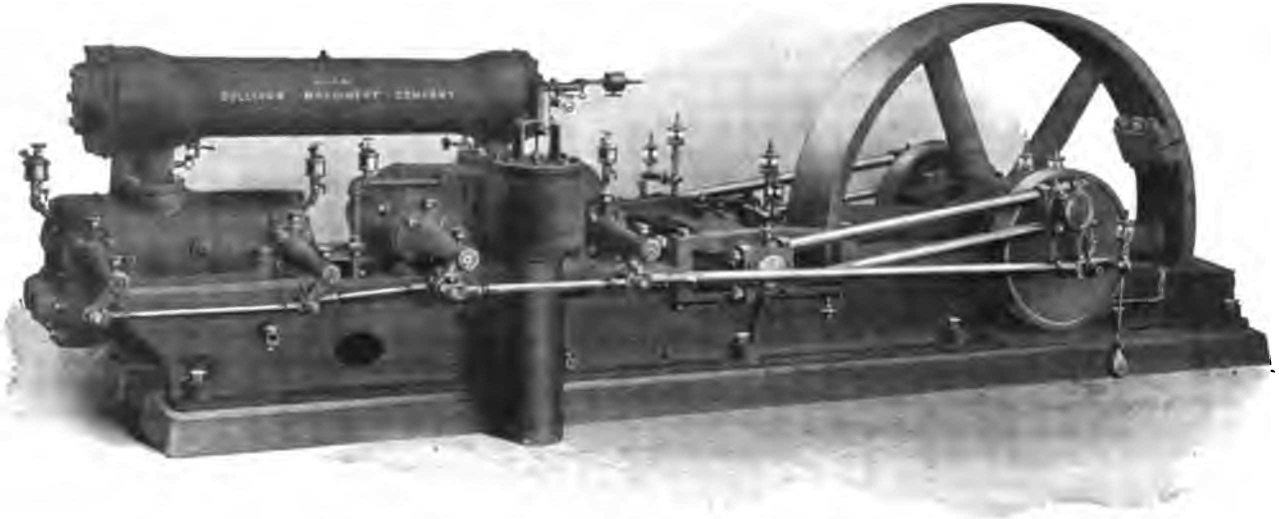


Fig. 61. Class WH-2 Straight-Line Two-Stage Belt-Driven Air-Compressor.
Equipped with mechanically operated Corliss air-inlet valves and removable poppet discharge valves. Air cylinder water-jacketed.
Sullivan Machinery Co., Chicago, Ill.

**For pressures of 500 to 1,500 lbs., three- or four-stage compression, preferably with duplex construction.**

**For pressures of from 1,500 to 3,000 lbs., or even higher, four-stage compression.**

A three-stage straight-line machine should be used only for small or moderate capacities. Beyond that, the duplex type is to be preferred because of its balanced construction and of the readiness with which it lends itself to compounding. Whether three or four stages shall be used for 500 to 1,500 lbs. pressure, is largely a matter of preference with different builders; but since four air cylinders are required for the duplex in either case (two intermediate cylinders being used in duplex three-stage work), probably the four-stage cycle is to be preferred for from 1,000 to 3,000 lbs. pressure.

The primary object of compound compression is a saving of power by eliminating what may be called the *heat element* in the total power required for compression. Other gains by compounding will be discussed later; but at this point it is to be noted that successful stage compression depends primarily upon the efficiency of the cooling devices used. It has already been shown that a water jacket is essential, but hardly to be considered as a great factor in the cooling process. The main burden of cooling, therefore, falls upon the device employed for cooling the air between compressions. This device is called the *intercooler*. Before taking up this feature, however, the gains by various degrees of compound compression will be investigated in a specific case.

Let the problem be to determine the best means of compressing 1,000 cubic feet of free air at sea-level to 100 lbs. gauge pressure. By reference to Table VII (page 27), it is found that, if this could be done isothermally, the power required would be 1,000 × .1317, or 131.7 H.P. The power required for this duty in single-stage adiabatic compression is 180 H.P.; for two-stage adiabatic compression, 154 H.P.; for three-stage adiabatic, 146 H.P.; and for four-stage adiabatic, 142 H.P. It is to be remembered that the values in this table assume adiabatic compression in the cylinders, but perfect cooling to initial temperature in the intercooler. Neither assumption is correct in practice, but the relative power requirements in the several cases are fairly shown by the values given. As compared with ideal isothermal compression, therefore, the efficiency of average single-stage compression in this case is the ratio of 131.7 to 180, or 73.2 per cent; of two-stage compression, 85.5 per cent; of three-stage compression, 90.2 per cent; and of compression in four stages, 92.7 per cent. At first glance, four-stage compression would seem to get first choice, three-stage second choice, and two-stage third choice. As a matter of fact, however, two-stage compression would probably be used; for while in theory there is undoubtedly better economy in three- and four-stage compression, yet in actual practice it is found that the added complication of design in the two latter cases, owing to the greater number of cylinders, valves, and working parts, with additional intercoolers, produces losses probably offsetting any gain by higher compounding. The limiting pressures for various stages of compression represent averages determined by the long experience of builders of compressors, and may therefore be safely accepted.

**The Intercooler.** The ideal intercooler is one which, at the full rated speed of the compressor of which it is a part, reduces the temperature of the air passing through it to the temperature of the cooling water which it uses. This probably can never be fully realized; but a good, practical standard for an intercooler is that it shall deliver air at a temperature not more than ten degrees above that of the cooling water. If the water is cold enough, this will bring the air down to, or even below, atmospheric temperature; and it is just as important that cool air shall enter the high-pressure cylinder as the low-pressure cylinder, for the reasons already given in considering the intake air temperature.

The successful intercooler, considered now as a heat-removing device, involves four fundamental requirements:

1. It must provide for a complete and minute subdivision of the air volume passing through it, in order that the heat may be dissipated without reliance upon the heat-conducting properties of the air itself. In other words, the air should be split up into thin sheets or streams so that the internal heat can be fully given up.
2. It must present an ample cooling surface to this subdivided air stream.
3. The circulation of the cooling water must be free and at such a velocity that the maximum amount of heat can be carried away. Furthermore, the flow of the water should be properly directed in relation to the flow of the air.
4. An ample cross-section must be provided in all air-passages, in order that the rate of flow of the air may be slow enough to give a length of contact with the cooling surfaces sufficient for the complete removal of the heat.

The first two requirements are met by the same means—namely, by the use of a nest of tubes inside the intercooler casing or shell, through which tubes the water circulates, and over, about, and between which the heated air flows. Baffle-plates should be so disposed as to cause the air stream to pass back and forth and between the tubes, splitting the air up into thin sheets. The prevailing practice puts the water inside the tubes, not the air; for in the latter case each tube would contain a column of air of which only the outer layer or film would be cool, the interior retaining its heat. This general arrangement gives the maximum cooling from a given flow of water.

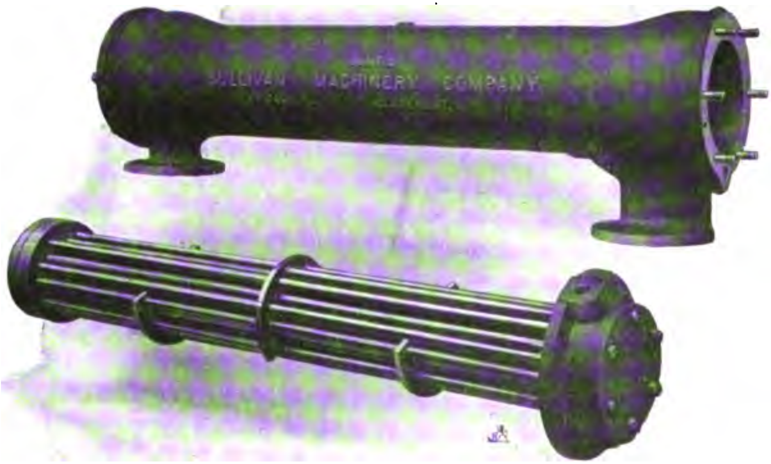


**Fig. 62.** Intercooler for Two-Stage Straight-Line Belt-Driven Air-Compressors.
Views showing shell, and tubes removed from same.
Sullivan Machinery Co., Chicago, Ill.

The third requirement is met by providing plenty of tubes giving an ample flow with a low velocity of the water; and by making the water flow through the intercooler in a direction contrary to that of the flow of air. The hottest air thus encounters the warmest water, and the air as cooled steadily comes in contact with cooler water.

The fourth requirement can evidently be met by making the intercooler shell large enough, and the tubes sufficiently numerous to provide an ample passage for the air without loss of pressure due to friction and without an undue velocity due to restricted area of passage.

One consideration of a purely mechanical nature, in the intercooler, is nevertheless as important as those just mentioned in its bearing on intercooler efficiency. Expansion and contraction must be provided for; otherwise there results either a leakage of air, with a loss of capacity and a waste of power, or a leakage of water into the air chamber, which may be carried over to the next cylinder, with disastrous results.

This suggests also another important point—namely, the proper drainage of the intercooler. The lowering of the air temperature condenses the water vapor in the air, and this water accumulates in the intercooler until it may cover some of the tubes, reducing the air passage as well as the cooling area. But the greatest danger is that this water, if not removed from the intercooler through the drain, will be caught in the sweep of air and carried in a mass into the next cylinder, when a broken cylinder-head or piston, or a bent piston-rod, may result.

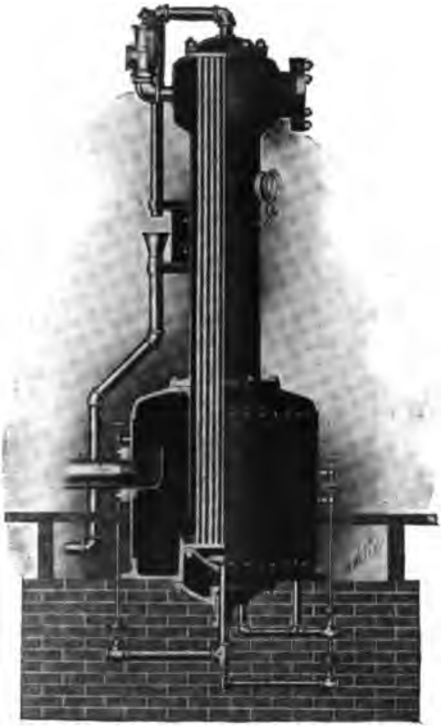


Fig. 68. Vertical Intercooler for Duplex Compressors.
Also used as an aftercooler.
Ingersoll-Rand Co., New York City.

**Other Advantages of Compound Compression.** The mechanical structure of an air-compressor must be equal to the maximum stresses which may come upon it; and this determines the amount and disposition of materials in it. The endurance of the machine depends upon its factor of safety; and good design exacts that this factor shall be large without extravagant use of metal. In other words, while the maximum stresses must be provided for, the average stresses must be kept well below this maximum if the machine is to operate satisfactorily in long, steady service.

Compound air-compression reduces the average structural stresses in the compressor, resulting in a higher factor of safety and longer life of the machine, with a given weight of metal. It has been seen that in a single-stage machine the piston starts against no resistance (except friction), but rapidly builds up pressure to the maximum, which is held until end of stroke. A 24-inch piston compressing to 100 lbs. gauge, single-stage, meets a total resistance of 45,239 lbs., or nearly 23 tons; and at 90 R.P.M. this is encountered and relieved 180 times each minute. It takes the best of metal, and plenty of it, to stand up under such service as this. But if this same work is done in two cylinders, with two stages, the condition is very different. The maximum total pressure on the low-pressure piston will depend upon the cylinder ratios; and the pressures in high- and low-pressure cylinders will partially balance and equalize. As a matter of fact, without going into the mathematics of it, it is found that the maximum stresses encountered in compression in two stages to a given pressure are only 55 or 60 per cent of what they would be in single-stage compression to the same pressure. In the case just cited, this means a reduction of terminal strains from 180 × 23 or 4,140 *ton-minutes* to 2,277 or 2,484 *ton-minutes.*

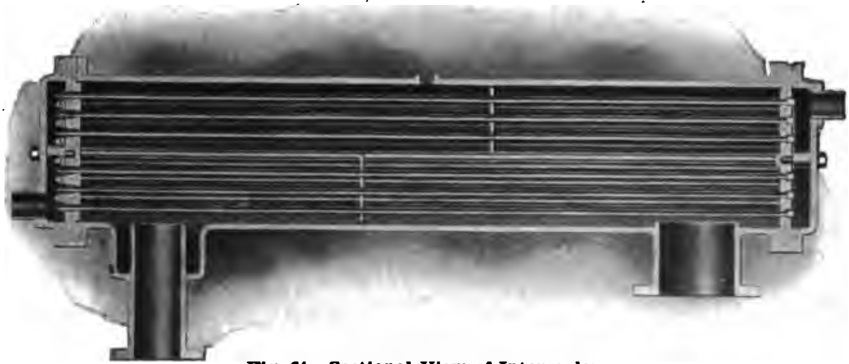
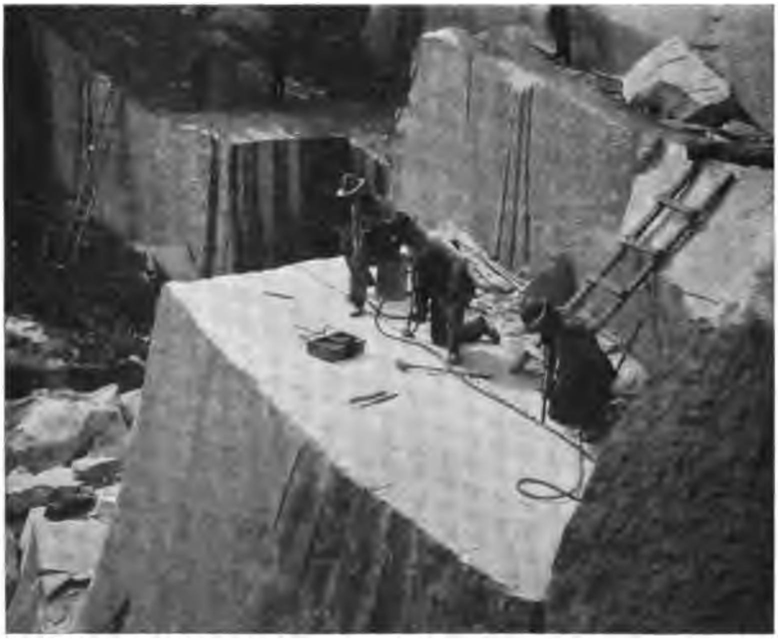


Fig. 64. Sectional View of Intercooler.
Laidlaw-Dunn-Gordon Company, Cincinnati, Ohio.

This difference in maximum pressure is found not only in the machine structure, but also in the air-valves, which are the most



**SULLIVAN PNEUMATIC PLUG**
Hammer Drills in Vermont Quarry

vital parts of a machine and which show a notably easier operation under these improved conditions.

But a very vital point must not be overlooked. Though the maximum stresses of average operation may be reduced and brought well within the limit of safety, yet the maximum possible stresses must be provided for in the design. For the failure of the supply of cooling water, preventing the cooling between stages, or the breaking of a high-pressure valve, may at any moment throw the full high-pressure load on the low-pressure piston, in which case disaster follows if the machine has not been designed with such a contingency in view. It is not safe to assume, therefore, that a multi-stage machine can safely be built lighter than a single-stage machine of equivalent capacity.

Another advantage of compound compression lies in the resultant improved steam economy if the compressor is steam-driven. This may be best illustrated by reference to the example just cited, where it was seen that the terminal stresses on the machine, and therefore the terminal power required, are reduced 40 to 45 per cent by compounding the air end. It was seen, also, that the load was more equably distributed throughout the stroke. Evidently this means a lower M.E.P. in the steam cylinder to do the work, secured by a shorter cut-off and consequently less steam. More of the work of passing dead center can be intrusted to the fly-wheel, instead of being provided for by a later cut-off. The mechanical gain by reduced terminal stresses also permits a higher piston speed with safety, this again improving the steam economy by reducing condensation losses and the possibility of leakage in the steam cylinder.

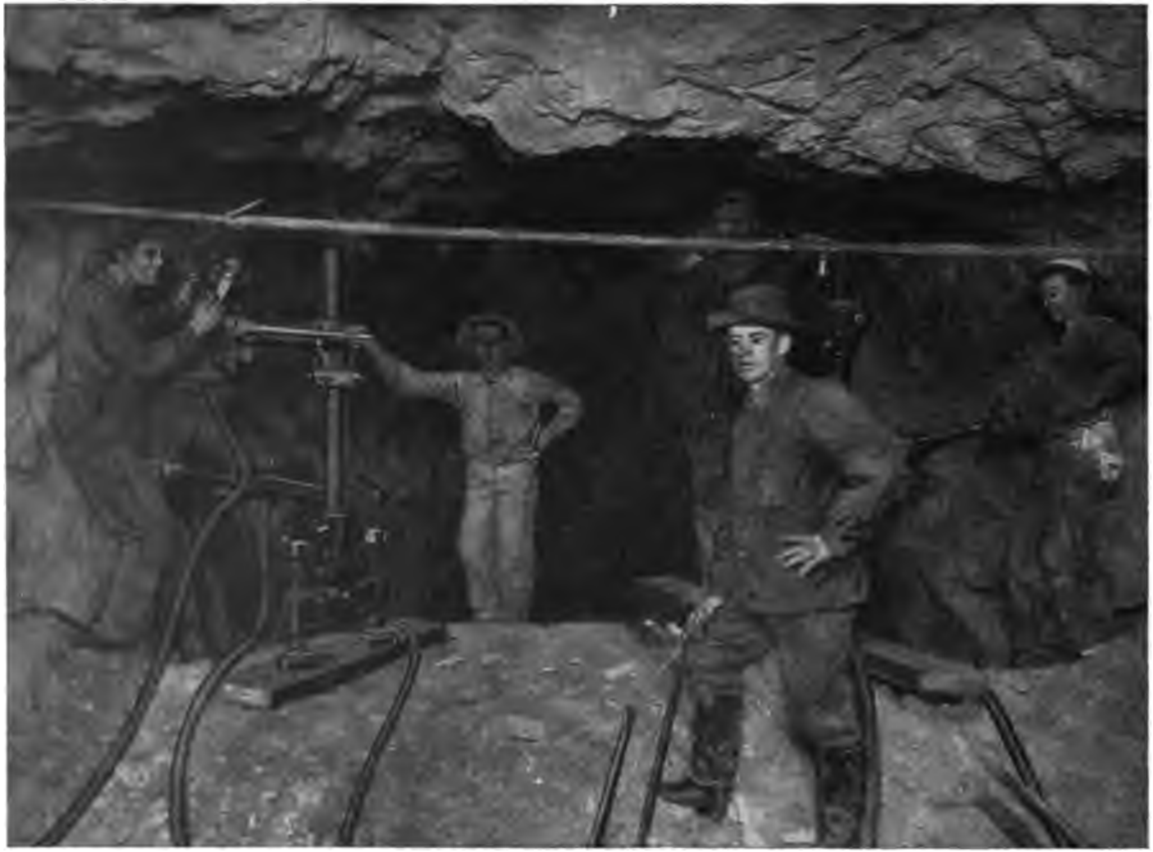
Higher volumetric efficiency comes with the use of compound compression, a gain in this respect being the result of three different causes. The first of these is the lower clearance volume; and it is to be remembered that, as affecting the free-air capacity of the compressor, the clearance in only the low-pressure or intake cylinder is involved. The air in the low-pressure clearance, at the end of the stroke, is at maximum pressure in that cylinder; and before free air can enter, this clearance air must re-expand to atmosphere. In a single-stage compressor, clearance pressure is the full terminal pressure of the machine. If this is (say) 7 atmospheres, and the clearance percentage is 1.5 per cent, the space occupied by this expanded air will be 7 × 1.5, or 10.5 per cent of the cylinder volume. If, however, the compression is to 7 atmospheres in two stages, the terminal pressure in the low-pressure cylinder will be somewhere around 3 atmospheres. In this case, therefore, the reduction in cylinder capacity due to the expansion of the clearance air (the percentage of clearance in the low-pressure cylinder being assumed the same as before) will be 3 × 1.5, or 4.5 per cent. The volumetric gain in this respect alone by compounding becomes evident.

The second cause of gain in volumetric efficiency by compounding, is the lower maximum temperatures in the cylinders, corresponding to the lower pressures. These being much lower in a compound machine than in a single-stage compressor, the pistons, cylinder walls and heads, valves, and ports are kept much cooler, and the entering air is not so much heated. This means a greater density of air at the beginning of compression, with a corresponding increase in volumetric capacity.

The third element in volumetric gain by compounding is the reduced leakage past valves, pistons, and rods in the compound machine, consequent upon the reduced extremes of pressure in each cylinder.

Compound compression results in more efficient lubrication of the machine, meaning easier running, improved mechanical efficiency, less wear, lower repair costs, reduced leakage losses, and longer life. The lower maximum of temperature which holds in each compound cylinder is not sufficient to affect seriously a good oil, and all parts are thoroughly and freely lubricated.

The last advantage of compounding is in the delivery of drier air. It was noted that a properly designed intercooler would condense and remove much of the moisture in the air. This reduces the liability to freezing-up at the exhaust ports of machines using the compressed air. It also largely obviates the accumulation of water in pipe-lines, where the latter are long, reducing their cross-section or even choking them entirely at low points.



COMPRESSED AIR ROCK DRILLS IN THE GUNNISON TUNNEL OF THE UNITED STATES RECLAMATION SERVICE, COLORADO
Sullivan Machinery Co., Chicago, Ill.

# COMPRESSED AIR

## PART II

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### AIR-COMPRESSOR VALVES

Two sets of air-valves are essential on each air cylinder of a compressor. The first are the *inlet valves,* their function being the admission of air to the cylinders; the second are the *discharge* or *outlet valves,* which govern the discharge of the air compressed.

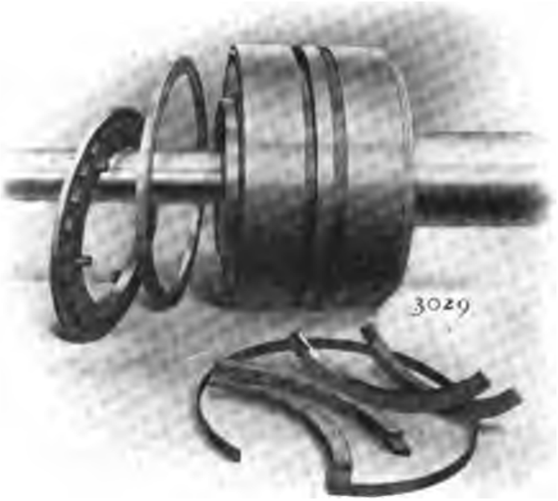

**Fig. 65.** Details of "Hurricane" Piston-Inlet Air-Valve.
Ingersoll-Rand Co., New York City.

While the number of different types of air-valves is as great as the number of compressor builders, yet an analysis of all types will show a broad classification into two groups: *Mechanically actuated* valves, which have a positive motion derived through some mechanical connection from the rotating or reciprocating parts of the compressor; and *poppet* or *automatic valves,* operated wholly by differences

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of pressure. There are some air-valves in use to-day which cannot be exactly placed in either class, notably the *piston inlet* valve of a prominent builder, which is operated by its own inertia. In still other cases the valve seems to combine the features of both classes, an automatic adjustability supplementing a primary mechanical control.

Mechanically actuated valves are frequently of the Corliss type, operated by levers from a wrist-plate driven by an eccentric or return-crank on the main shaft. Usually there is nothing in this valve movement corresponding to the release mechanism of the Corliss steam valve gear, since there is nothing in the air card corresponding exactly to the phenomenon of cut-off on the steam card. Another form of mechanical valve is similar in general appearance to the regular poppet valve, but its movement is mechanically produced. A third form of valve which may be considered in this class is the *air-thrown* poppet valve of one leading manufacturer, in which the movement

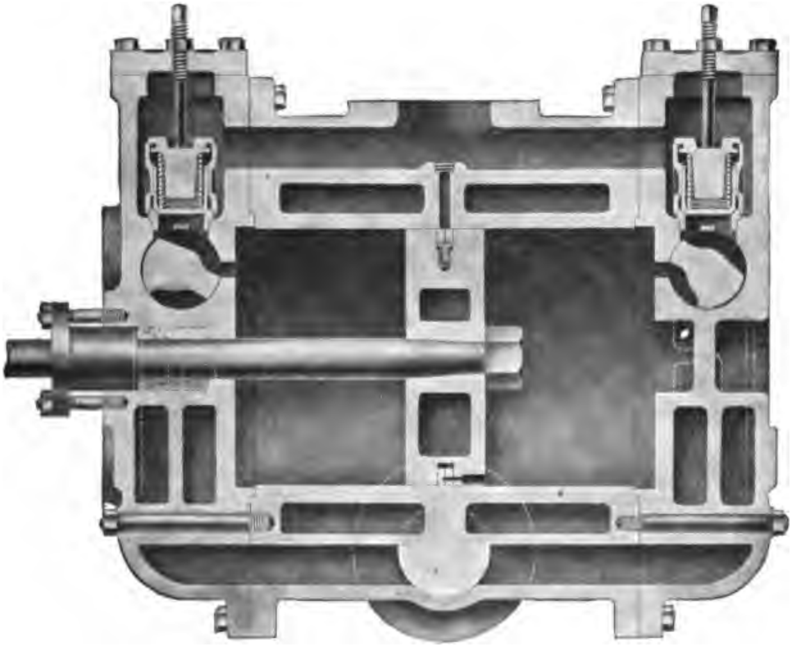

Fig. 66. Section of Air-Compressor Cylinder with "Cincinnati" Valve Gear for Inlet and Discharge, with Poppet Valves Supplementing the Corliss Valve on Discharge. Laidlaw-Dunn-Gordon Co., Cincinnati, Ohio.

of a poppet valve is produced by air-pressure from the receiver or intercooler on a small piston on the stem of the valve.

Mechanical air-valves are practically limited in their application to the inlet valves of single-stage compressors, or the low-pressure inlet valves of compound machines. There are instances, however, in large sizes, where mechanical valves are used on the intake of the high-pressure cylinder as well. Almost the only instance in which mechanical valves are used for discharge purposes is found in the case of some blowing engines, which are simply compressors designed for very low pressures. The function of an atmospheric inlet valve is simply to open full when the clearance air has expanded to atmosphere, and to remain open to the end of the stroke. This duty is light, for no pressure higher than atmospheric pressure is encountered while the valve is in motion. The same valve used for the high-pressure inlet of a compound machine, while subject to higher pressures—namely, the discharge pressure of the low-pressure cylinder—is nevertheless not compelled to move while under this pressure.

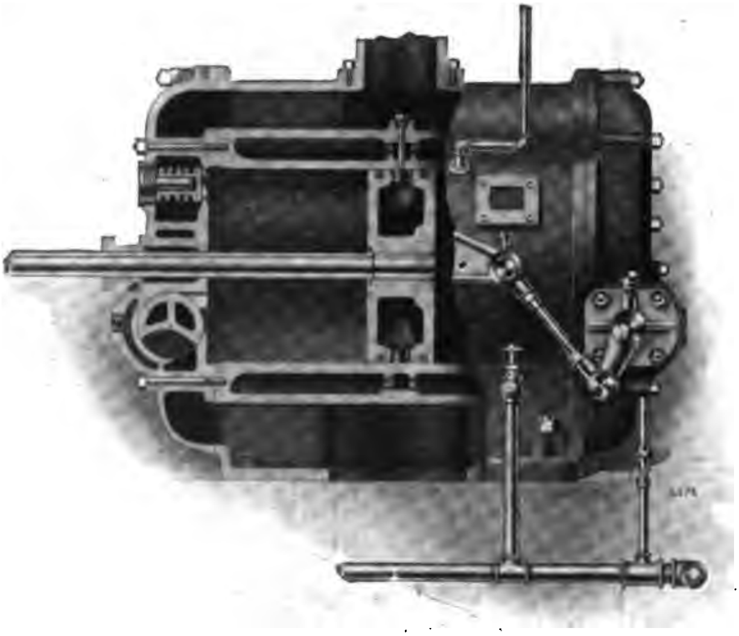

**Fig. 67. Section of Compressor Cylinder with Rand-Corliss Inlet Valves and Direct-Lift Poppet Discharge Valves. Ingersoll-Rand Co., New York City.**

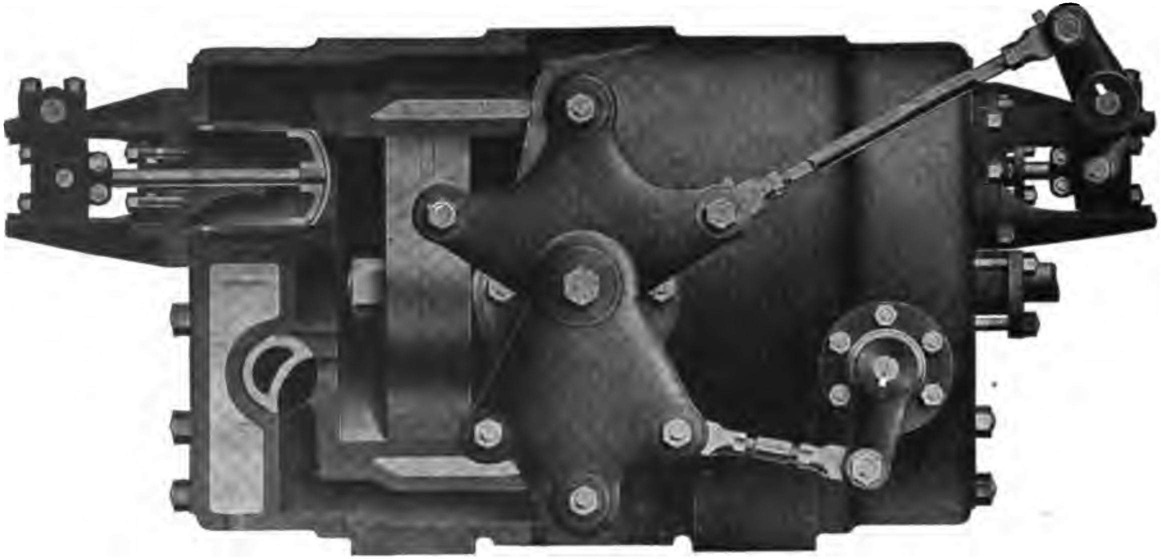

Fig. 63. Section of Compressor Cylinder with Corliss Inlet Valves and Mechanically Operated Poppet Discharge Valves. Allis-Chalmers Company, Milwaukee, Wis.

However, when used for controlling the air discharge, mechanical valves suffer under the necessity of opening at a fixed point of the stroke, while the correct point of discharge will vary in the stroke with varying receiver pressures, even where automatic pressure-regulators are employed. In this event, one of two things will happen.

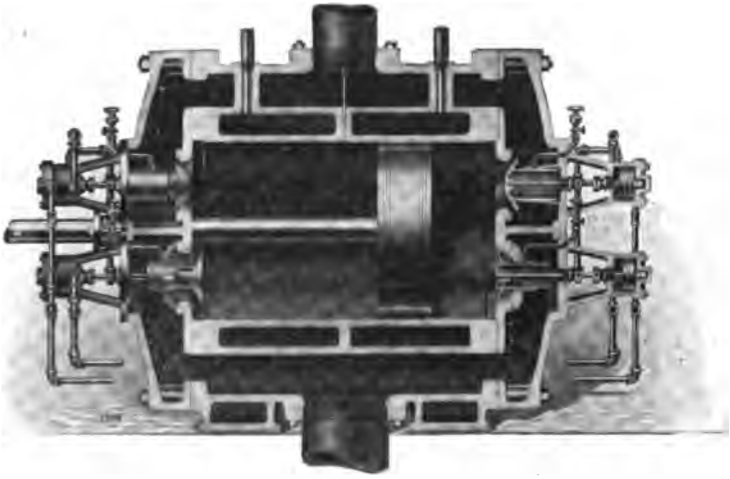

Fig. 69. Section of Compressor Cylinder with Air-Thrown Inlet and Discharge Valves. Ingersoll-Rand Co., New York City.

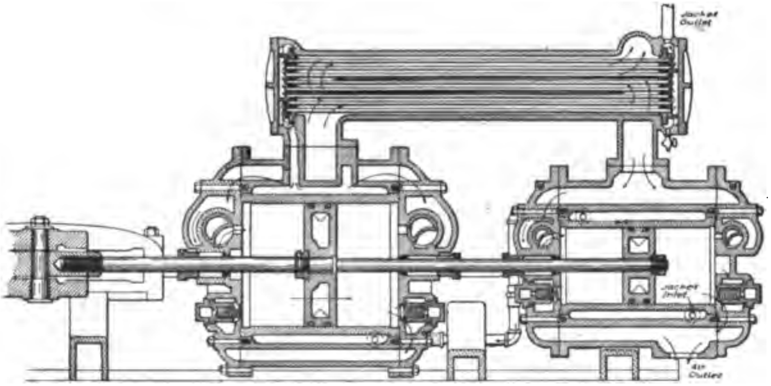

Fig. 70. Section through Inter-cooler and High- and Low-Pressure Cylinders of Sullivan Straight-Line Compressor. Corliss Inlet and Poppet Discharge Valves on Both Air Cylinders.

If the point of discharge comes *before* the point of discharge valve opening, due to a drop in receiver pressure, the valve and valve gear must move under a heavy, unbalanced load from the excess of cylinder pressure over receiver pressure. If the point of discharge falls *after* the point of valve opening, the mechanism is again under heavy strain when it moves, due to the excess receiver pressure. In either case the duty is hard on the valve and gear, wear is rapid, and leakage almost inevitable.

Poppet valves are simply held to their seats normally by a spring, and move when the air-pressure upon their face overcomes the spring tension. As applied to atmospheric inlet purposes they are open to the objection that sufficient vacuum must be produced within the cylinder on the admission stroke to give an excess of atmospheric pressure sufficient to overcome the spring tension. This brings the admission line below atmosphere, and reduces the volumetric efficiency. This objection, of course, does not hold on the high-pressure inlet of compound compressors, where the pressure is all that is required for the prompt opening and closing of the valve. The poppet type is very generally used for high-pressure inlet purposes, on this account.

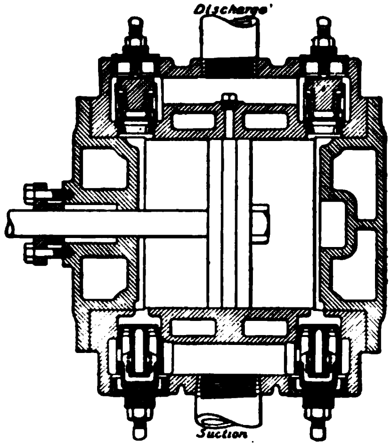

**Fig. 71.** Section of Air Cylinder of Laidlaw-Dunn-Gordon "Climax" Compressor, with Poppet Inlet and Discharge Valves.

The poppet valve is also almost universally applied for discharge purposes, on both high- and low-pressure cylinders. In this service, there is plenty of pressure to operate such a valve against its spring; and since it will not open until the relation of pressures on its front and back is exactly what it should be, it automatically adjusts itself to varying discharge pressures.

**Air-Inlet Valves.** Without going into a detailed discussion of the many different types of air-inlet valves, some general observations may be made as to the functions and requirements of these important parts. The functions of the air-inlet valve (or valves) are: To admit the maximum volume of cool, clean air to the cylinder, with the least loss or expenditure of power; to keep this air in; and to continue to perform this duty indefinitely, at the least possible cost. An analysis of these fundamental facts reveals the following requirements of a successful air-inlet valve:

1. It must admit a volume of air which will entirely fill the compressor, this air being as nearly as possible at atmospheric pressure. This calls for an instantaneous and complete opening of the valve, so timed that there shall be no escape of the compressed air in the clearance space. This full opening should be held to the end of the stroke. This first consideration calls further for short, direct, and unobstructed passages through the ports and valves.

2. It must admit clean air, suggesting a design permitting ready connection to an intake passage or conduit supplying air from a place free from dust and grit.

3. The air admitted must be as cool as possible. This is partially met by the second consideration, whereby cool air as well as clean air should be supplied; but, more than this, it demands that this cool air supplied shall nowhere be heated in its course into the cylinder, suggesting that valves, ports, and air-passages should be so placed as to be cooled by the cylinder jacket. Moreover, the air should be admitted through one or only a few large openings, rather than sieved through many small ones.

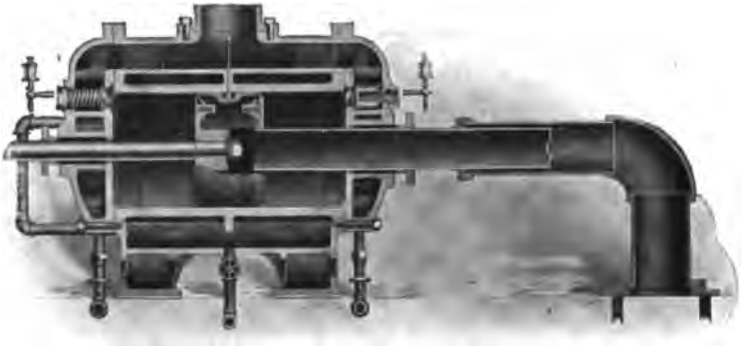

Fig. 72. Section of Compressor Cylinder with Hurricane-Inlet and Direct-Lift Poppet Discharge Valves.
Ingersoll-Rand Co., New York City.

4. All the air admitted must be kept in the cylinder, calling for instantaneous and complete closing of the valve at the end of the stroke, with a perfect seating and with no leakage.

5. All the processes hitherto outlined must be carried on with the least possible expenditure of power for the manipulation of the valve gear and the overcoming of friction—a condition necessitating nice adjustments, simplicity of mechanism, perfect lubrication, and as few bearing and wearing surfaces as possible. This exacting performance must be continued indefinitely, twice each revolution and hundreds of times per minute, without undue wear or loss of adjustment. This condition is fundamentally one of correct design, high-class materials and workmanship, and generous lubrication.

The ideal inlet valve has here been outlined. Probably no compressor on the market to-day exactly fulfils all these conditions, but many types represent the judgment of their several builders applied in a wise compromise. In the foregoing statement of primary essentials, it is to be remembered that these apply to high-pressure as well as to low-pressure valves, and they can be studied in relation to high-pressure work by substituting *intercooler pressure* for atmospheric pressure wherever the latter has been used in defining these essentials.

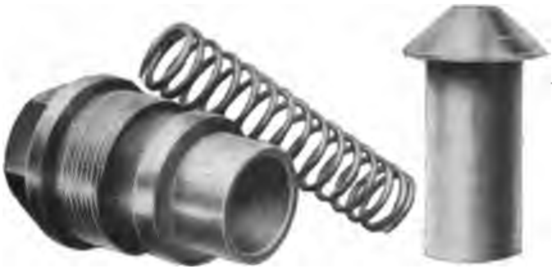

Fig. 78. Details of Poppet Discharge Valve. Blaisdell Machinery Co., Bradford, Pa.

**Air-Discharge Valves.** The functions of the air-discharge valve (or valves) of an air-compressor are: To release the full volume of compressed air from the cylinder, with the least power expenditure; to keep it all out after discharge; and to continue indefinitely, without undue expense, to maintain this duty. The primary essentials of such a valve are as follows:

1. The opening of the valve must be full and instantaneous, occurring at just the right point in the stroke; and this opening must be held until the piston comes to rest. The valve opening must be ample, so that the velocity of discharge may not be excessive; and the ports and discharge passages should be short and free from obstruction.
2. As few parts as possible—and these correctly adjusted—should provide for the necessary valve movement, with as little power consumed in friction as possible.
3. The valve should close instantly and completely as soon as the piston comes to rest at the end of the stroke, in order that there may be no return of the air discharged when the piston starts on its back-stroke. Such a return of compressed air would act as an increased clearance to reduce the capacity of the cylinder.
4. The continued maintenance of this performance—more arduous upon the discharge valves than upon the inlet valves, because of the higher pressures encountered—suggests the advisability of a simple construction with as few bearings as possible, with careful workmanship, the use of good materials, complete lubrication, and a careful distribution of materials to give the best results.

It is always to be remembered that air-discharge valves and ports are continually exposed to a flow of heated air, even where the most careful jacketing is used. On the other hand, a requirement of inlet valves and ports is that they should be kept cool. The latter should therefore be as far removed as possible from the former; nor should the same ports be used for both inlet and discharge.

**Areas of Valves, Ports, and Passages.** It might at first glance seem that the area of the discharge valves, ports, and passages could be much less than the corresponding inlet areas, since the former must pass air at a higher pressure and therefore in less volume than the latter; but it is to be remembered that while the inlet areas are open during the full stroke, the discharge areas are open for only a part of the stroke—usually about one-third. In order, therefore, to make the velocity of discharge about the same as that of intake or admission, it is the general practice to make the inlet and discharge areas approximately equal. It is important that the velocity of flow through valves, ports, etc., shall not be excessive; and this depends upon the piston speed, which is a widely variable quantity. The higher the piston speed, the larger should be the valve and port areas. In actual practice, the inlet area varies from 3 to 15 per cent of the cylinder area, with an average probably around 9 or 10 per cent. The corresponding discharge areas run from 7 to 15 per cent, with an average in the neighborhood of 10 or 12 per cent. Owing to the increased friction through a number of small passages, as compared with that through a single large one, the actual *measured* area in the former case should be 50 to 100 per cent larger than in the latter. This explains why the measured inlet area of machines with Corliss or other large-area inlet valves is so often much less than the measured discharge area of many small poppet discharge valves, while still giving good results.

**Inlet and Discharge Port Areas.** While reference has been repeatedly made to the clearance of an air-compressor, it may be well to recall at this point that clearance is really made up of three elements: (1) the *inlet port clearance,* which is all the space left between the closed inlet valve and the inner surface of the cylinder-head; (2) the *discharge port clearance,* consisting of all the space between the closed discharge valves and the inner surface of the cylinder-head or walls; (3) the *piston clearance,* which is the space allowed for safety between piston and cylinder-head at the end of the stroke. The latter element is an essential in all compressors; but it is not unusual for the other two elements together to constitute by far the larger proportion of the total clearance. This suggests the superiority of those valve designs in which inlet and discharge port clearances are reduced to the lowest practical limit.

## MECHANICAL EFFICIENCY OF AIR-COMPRESSORS

It will be remembered (see page 32) that an air-compressor can be considered as made up of three elements—a driving element, a compressing element, and a power-transmitting element, which latter constitutes the mechanical structure of the machine. The first two have been discussed; the last remains for consideration.

It has been seen that the mechanical efficiency of a compressor—the ratio of power output to power input, which is the ratio of the I.H.P. of the air cylinder to the I.H.P. of the steam cylinder—can be exactly determined by a comparison of the steam and air indicator cards. The difference between the I.H.P.'s of these two cards represents power consumed in the friction of the machine. Good mechanical efficiency, therefore, is seen to depend upon the extent to which friction is reduced. No attempt will here be made to enter into the details of compressor design. It will be enough simply to enumerate the points which have a vital bearing on this subject:

1. Power should be applied to resistance—steam pressure to air pressure—in as direct a manner as possible, so that bending or deflecting strains may be avoided as far as possible, with the resultant binding of bearings and loss of correct alignment.

2. Ample metal should be used, properly disposed to afford the rigidity necessary as a further precaution against the attendant binding stresses just mentioned. A light construction is always to be avoided.

3. Lest this correct proportioning of materials to meet stresses should result in an excessive weight and an extravagant cost, these materials should be of the best quality and intelligently selected for specific purposes, affording the maximum strength per unit of weight.

4. All bearings should be of ample area, so proportioned as to give a safe, moderate pressure per inch of bearing surface.

5. Wherever possible, bearings should be supplied with removable linings (preferably of anti-friction metal). In all cases, whether thus lined or otherwise, they should be scraped smooth and true, properly adjusted and arranged for the ready distribution of lubricant—all of this tending toward a lower coefficient of friction for these bearing surfaces.

6 Complete provision should be made for a plentiful supply of lubri-

cant to all bearing surfaces, furnished preferably by an automatic system which will be effective and efficient at all speeds.

7. The utmost simplicity should be sought throughout, reducing the number of parts, the number of joints (each an element of weakness), and the number of bearings, with their attendant friction and their possible loss of economical adjustment.

8. The design should afford a ready accessibility at every point, inviting frequent and careful attention to the maintaining of correct adjustment of all parts for most efficient operation.

9. Steam compounding and compound air-compression have important effects upon the mechanical efficiency, which have already been discussed in the proper place.

Friction has been aptly defined as "that portion of the power consumed by a machine in wearing itself out." This definition throws on the question of compressor endurance a light in which the importance of a high mechanical efficiency appears in bold relief. True economy is a question, not of momentary results, but of results covering months and years of continuous service under all conditions; and a compressor in which the percentage of "wearing-out power" is least promises not only the highest, but also the longest, record of sustained economy and satisfactory service.

## IGNITIONS AND EXPLOSIONS

Ignitions and explosions in compressor cylinders, air-receivers, and pipe-lines, while comparatively rare, are yet of sufficient frequency to warrant attention. Such instances evidence a condition which should not exist in a well-designed and properly managed air-power plant; and several causes may account for them. In all cases it is to be remembered that air itself is not inflammable; and combustion or explosion can occur only when a foreign inflammable substance is present in the air. This is usually an explosive vapor produced by the volatilization of the lubricant used in the compressor; and ignition or explosion is always the result of some abnormal increase in temperature above the flash-point of the oil.

A common mistake in the operation of air-compressors is to use more oil than is necessary in the air cylinders. In steam cylinders more oil is required, because the tendency of the steam is to cut or wash away the oil; but in air cylinders there is no such action, and a drop of oil now and then is all that is necessary. Where too much oil is used, there is a gradual accumulation which may clog the valves and passages, interfering with the valve action and air discharge, and contributing toward a dangerously high temperature. Only the best oil—*never cylinder oil*—should be used in the air cylinders.

Benzine, naphtha, kerosene, and other light oils should never under any circumstances be introduced into a compressor cylinder. While they are unquestionably good for cleaning the cylinder walls, valves, ports, etc., they are an element of great danger, because of the ease with which they are volatilized and ignited. Soap and water is practically as good for cleaning purposes, and there is no danger attendant upon its use. A good practice is to fill the cylinder lubricator with strong soap and water once or twice a week—or even oftener, if there is a pronounced tendency to clog up—feeding it into the cylinder just as oil is fed.

Several causes may contribute toward the higher temperatures necessary for ignition and explosion. There may be an excessive temperature produced by an abnormal pressure not shown by the gauge, or there may be a very high temperature produced without any increase in pressure.

In the first case a poor oil may cause deposits of carbon in ports, passages, pipes, etc., which will so reduce the discharge area that a momentary pressure sufficient to produce an ignition temperature may be reached, even though this pressure may not extend to the receiver and be shown there on the gauge. Insufficient valve and port area may accentuate this trouble, and numerous bends in the pipe between the compressor and cylinder may still further contribute. In the second case, if a discharge valve should stick and remain open because of the carbon deposit upon it, the heated discharge air may return and be mingled with the free air entering. This air, already at a high temperature, would then be compressed with a further heating, and the final temperature might well be above the flash-point of the oil used, though the pressure might not exceed normal.

Heated intake air may also produce a dangerously high discharge temperature in the same manner as that just described. For instance, if the air for the compressor is drawn from a heated engine-room at a temperature of 80 or 90 degrees, its final temperature on compression may easily be above the danger point.

Explosions are much less likely to occur with compound compressors having intercoolers; and an aftercooler is an additional safeguard. Freedom from explosion and ignition can be practically assured by the following means: By providing for the admission of cool intake air; by the use of only the best oils, in the minimum quantity possible; by never using kerosene, benzine, naphtha, gasoline, etc., in the air cylinders; by frequent inspection and cleaning of air-valves, ports, cylinder bore, etc.; by the use of an aftercooler; by making the pipes between compressor and receiver of large area and as short and free from bends as possible; and by selecting a compressor with ample valve and port areas and with direct air-passages.

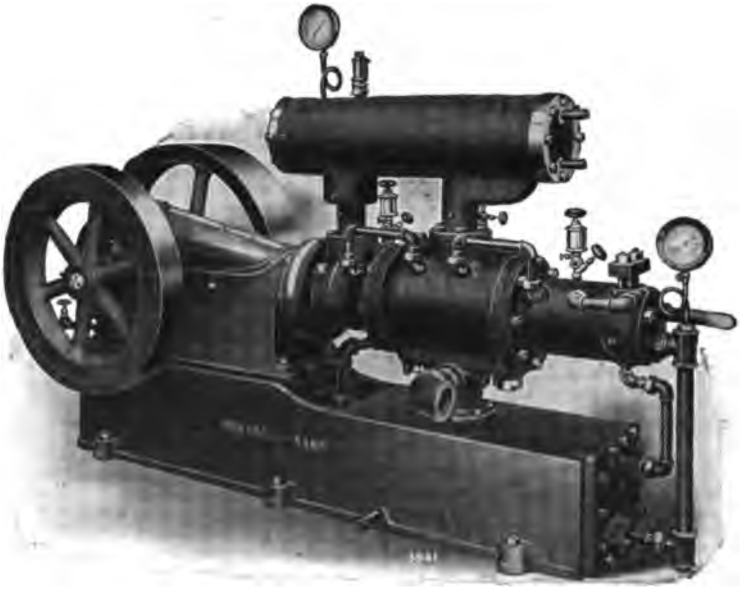

Fig. 74. Class EE-3 Straight-Line Three-Stage Belt-Driven Compressor for High Pressures.
Ingersoll-Rand Co., New York City.

## HIGH-PRESSURE COMPRESSORS

The development of pneumatic haulage requiring the storage of air at high pressure, the use of storage air-brake systems, torpedo work, etc., have developed the modern high-pressure compressor for pressures up to 3,000 lbs. These are essentially multi-stage machines, and may be straight-line or duplex in type. The requirements of this line of work are particularly exacting, because of the high pres-

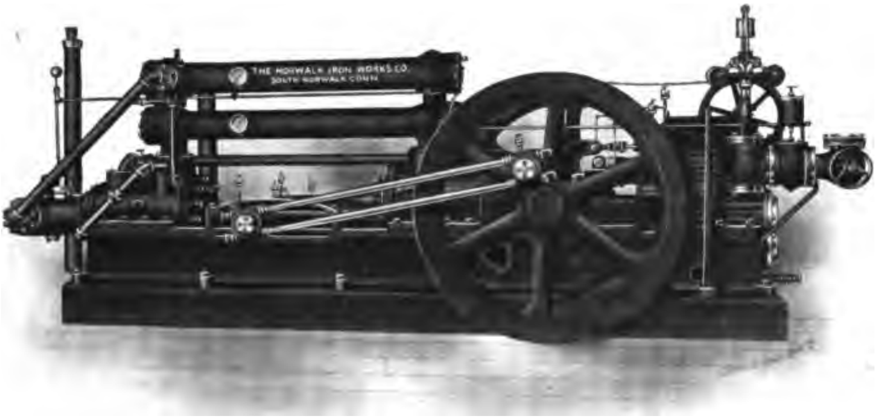

**Fig. 75.** Straight-Line Three-Stage Compressor for High-Pressure Locomotive Charging. Norwalk Iron Works Company, South Norwalk, Conn.

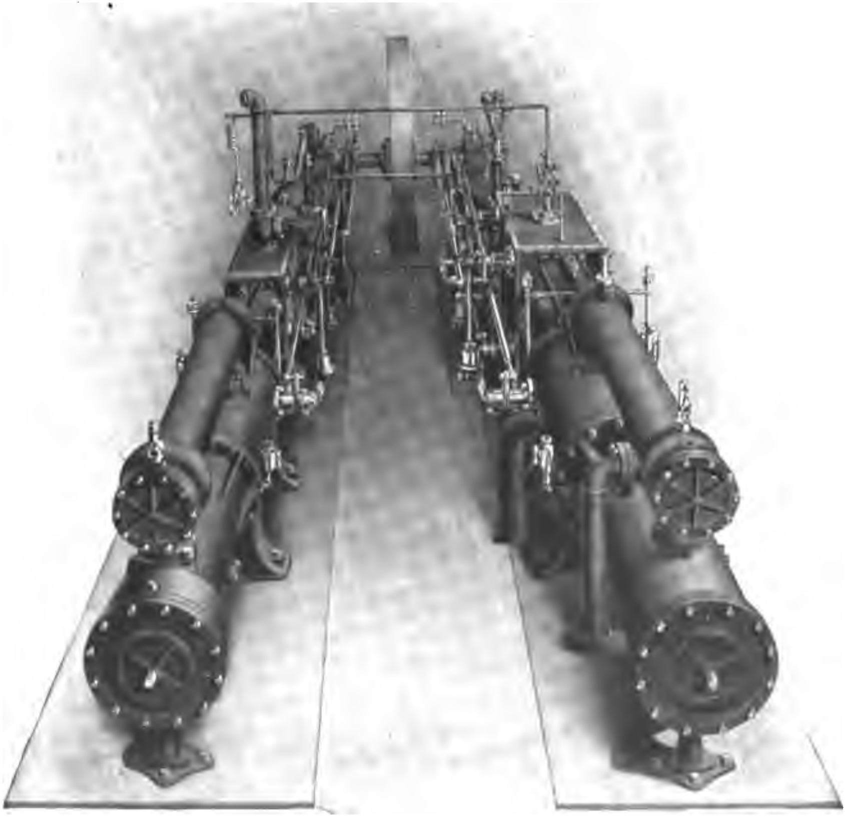

**Fig. 76.** Class WX-4 Cross-Compound Steam and Four-Stage Air-Compressor. Sullivan Machinery Co., Chicago, Ill.

sures involved; and the greatest care in design, workmanship, and materials of construction is necessitated.

The greatest difficulty is in the handling of the high temperatures which may be encountered, and the necessity for very perfect and complete cooling devices. It will be evident that the ordinary tubular intercooler cannot always be used for this work in the higher stages, because the volume to be handled becomes so small. The various high-pressure machines on the market, therefore, differ principally in the details of construction of intercoolers, high-pressure valves, high-pressure packings, cooling devices, etc. Practice varies so that no details can here be taken up without going into a discussion of several machines of specific builders; and this has hitherto

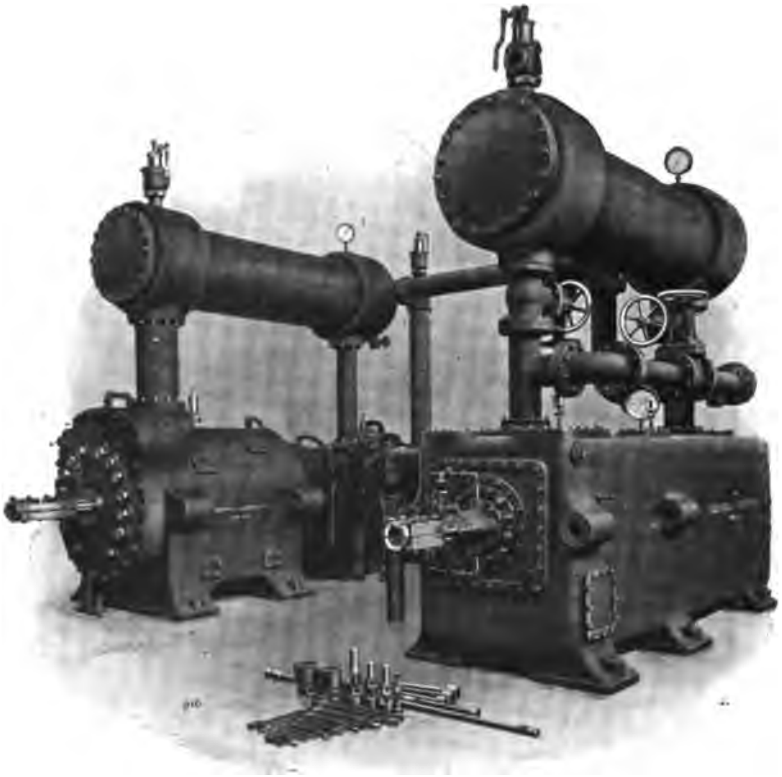

Fig. 77. Four-Stage Air Cylinders for Double Cross-Compound High-Pressure Air-Compressor.
Ingersoll-Rand Co., New York City.

been avoided in this treatise, which deals only with fundamental principles. The catalogues of the various builders may be referred to for more complete information on this point.

## GAS-COMPRESSORS

The modern gas-compressor differs but little from the standard air-compressor; and the general principles already laid down are equally applicable to the compression of air and gas. It is, of course, to be understood, that the physical properties of various gases differ from those of air, making necessary the statement that the tabulated data presented in these pages cannot be used for gas-compression. For instance, the exponent of the equation for adiabatic compression of gas differs from that of air; but the fundamental principles of design, stage compression, cooling, etc., apply to gas as well as to air. Indeed, with some gases, the question of cooling is even more important than with air as affecting the thermal or illuminating power of those gases.

Another peculiarity of gas compression is the necessity for an absolutely closed intake and for the elimination of all leakage, for the escape of gas may seriously affect the health of the attendants, or may even result in explosion where the conditions are suitable. Occasionally a gas must be handled which chemically affects the metal of the ordinary compressor, in which case the cylinder bore, piston, rods, valves, and other parts in contact with the gas must be made of a special metal to resist this action. Gas-compression is a subject for special study, and requires a broad experience and a more or less specialized training.

## DESIGN OF THE AIR-POWER PLANT

Compressed-air power plants may be broadly divided into two general classes. The first of these is the plant for temporary or semi-permanent use, which supplies power for a specific job, after the completion of which the plant may be sold or moved to another location. In this class will come the average plant for mining and contracting, for running rock-drills in tunnel work, operating pneumatic tools for bridge or structural work, for the development of a mining property, etc.

The second class includes the plant for permanent and continuous service, such as for running drills, coal-cutters, and pumps in a developed mine, operating a quarry, supplying air for railway switch and signal work, factory or municipal water supplies, railway and manufacturing shop service, and kindred work.

For work covered by the first class, the straight-line compressor is usually preferred, because of its self-contained character, its ease of transportation and installation, its simplicity, and the readiness with which it lends itself to the care of attendants of only ordinary skill. In this class of service, high operating economy is usually of less importance than the ability to "keep going" day after day, often for weeks at a time, without a shut-down. Frequently the work must be completed within a given time, or other considerations demand the utmost speed in the prosecution of the work. Under such conditions, it is of primary importance that the air-power plant shall be absolutely dependable, this point having even greater weight than the lowest steam consumption. Steam valves of plain slide or of Meyer cut-off type are usually preferred to more refined valve movements. Steam pressures are usually those which can be furnished by the ordinary return-tubular boiler, which can easily be transported and erected. There is, however, a notable tendency in recent years among the most successful contractors, to install high-grade duplex Corliss machines and water-tube boilers with high steam pressures and condensing apparatus, even for jobs which are more or less temporary in character. The saving in fuel and other operating charges by this advanced practice is evidently an important consideration in large enterprises where competition is keen and where ultimate profit depends upon close attention to every economy, however seemingly small.

In power plants of the second class, practice recommends the use of strictly high-grade compressors; and preference leans toward the double cross-compound type with Meyer cut-off or Corliss steam valves, the use of high steam pressures, a condensing equipment, and other accessories, making up a plant of the highest fuel and steam economy. Such plants are usually in charge of skilled engineers, and repair facilities are immediately at hand in emergency. It is not to be understood from this that the better class of compressors are more liable to breakdown. On the contrary, all things considered, they are fully up to the standard of the simpler machines; but any mechanical refinement adopted with a view to higher economy is likely to introduce elements requiring closer attention and demanding the maintaining of exactly correct adjustments. This is the price which must be paid for better efficiency, but it need not mean weakness or liability to breakdown. The very fact that a machine does better work, justifies its claim for better care and closer attention.

As to the boilers, steam piping, feed-water heaters, condensers, etc., involved in the high-grade air-power plant, no extended discussion is necessary here. The fundamental principles of steam-power plant design apply in this case.

As to the layout of the compressor plant proper, probably the first consideration is that of adequate protection against total loss of air supply by accident to the compressor installation. Obviously this lies in the subdivision of the plant into two or more units, in proportions to be determined by a careful study of the conditions under which the machines must work, the character of their load, etc.

It is always to be remembered that an air-compressor has no overload capacity. Its full rated speed as determined by the manufacturers is the limit of safe operation, and it can be exceeded only at a risk. The economy of the duplex steam-driven machine is practically constant at all loads up to full rating; and therefore nothing is lost in economy by having one compressor capable of carrying the full load, while still automatically regulating itself under fractional loads. But in such a case a breakdown of that machine would cripple the entire system. Moreover, provision must be made for occasional inspection and overhauling of the machine without stopping the operations depending upon it. It is safer, therefore, to have two compressors, each of one-half the maximum required capacity. One machine would then run at full capacity so long as the load on the plant did not fall below half of full load, the other compressor meantime caring for the intermediate fluctuations. For loads between half-load and no load, one machine would be shut down, saving oil, wear and tear, and attention. In case of accident to one machine, the other could continue to supply half the demand. Conditions may readily be conceived—and, indeed, are frequently met in practice—where three, four, or even eight duplex compressors afford the best solution of the problem of economical operation and regulation.

But where the compressors are power-driven—usually by electric motors—mere duplication of units is not sufficient, for the electric motor falls off in efficiency at fractional loads, and it is wise to keep all the compressors running as far as possible at full load. A theoretical case will best illustrate the method of handling such conditions. Let the problem be to install an electrically driven compressor plant to work under load variations of from 75 H.P: to 500 H.P., with an average normal load of 400 H.P. Probably the best solution would be the use of three motor-driven compressors, two of 200 H.P. and one of 100 H.P. The smaller unit would have a high-grade unloading device. Such a plant would afford the following combinations; 500, 400, 300, 200, and 100 H.P.—in all of which the machines in use would operate under full load and at maximum efficiency. When load fell below 100 H.P., the unloader of the smaller unit would safely care for a reduction of 25 or 30 per cent.

## TRANSMISSION OF COMPRESSED AIR

The compressed-air transmission and distribution system is here understood as including all apparatus taking the air from the compressor discharge and delivering it to the machine where it is applied. With this understanding, the transmission system will be seen to include the following: (1) the *aftercooler,* if one is used; (2) the *primary air-receiver* or *receivers*; (3) the *secondary receivers* which may be used along the line; (4) the *pipe-line* proper; (5) the *reheater,* if one is used. These several divisions will be taken up in order; but before their discussion, the general theory of compressed-air transmission will be considered.

It will be remembered that the discharge from any air-compressor is more or less heated; but the compressed air quickly gives up its heat whether an aftercooler is used or not; and thereafter the transmission of this air may be considered as isothermal, or at a constant temperature approximately the same as that of the surrounding atmosphere. The initial cooling of the air after discharge and on just entering the transmission system, represents, of course, a loss of power; but inasmuch as the transmission system proper is not re-

responsible for this loss, the latter cannot be charged against the transmitting devices.

With the air cooled to normal temperature in the system, and assuming for the present that there is no leakage, there can be no loss of power during transmission. A drop in pressure there will be, in an amount determined by the physical characteristics of the transmission system. But since the conditions are isothermal—that is, $PV$ = a constant—this drop in pressure is accompanied with a corresponding increase in volume, and the intrinsic energy of the air in transit remains practically the same.

The drop in pressure accompanying transmission (assuming that there is no leakage) is the result of friction in the pipe-line and accessories; and there is no known formula which adequately covers even average conditions. Authorities differ widely, and even the results of actual tests have revealed no basis for accurate computation. However, the drop in pressure, due to friction, depends upon the following factors: the volume of air; the pressure of the air; the velocity of the flow; the diameter of the pipe; the length of the line and the condition of its interior surface; the number and the angle of elbows or bends; and the number and style of valves used. Moreover, the velocity of flow is not uniform, for the steadily reducing pressure and the steadily increasing volume produce a gradual acceleration in velocity, in order that the larger volume may be passed by the pipe of fixed diameter.

It is considered best in this treatise, therefore, to omit any formulæ for compressed-air transmission, which, at the best, would be only approximate for any particular case.

**Tables of Air-Power Transmission.** The problem usually confronting a practical man dealing with compressed air is the selection of a pipe of suitable size which will deliver the output of a given compressor plant over a given distance, with a given initial pressure, and with a drop in pressure not to exceed a certain number of pounds which may be arbitrarily decided upon. It will often be useful, also, to know the equivalent volume of compressed air delivered under these conditions.

Tables XI, XII, XIII, and XIV, which are copyrighted by the Ingersoll-Rand Company and here used with their permission, give all the data above referred to for four common initial pressures—

# TABLE XI

## Loss of Pressure in Pounds, by Friction, in Transmission of Air through Pipes 1,000 Feet Long

Copyright, 1906, by Ingersoll-Rand Company

Initial Air-Pressure, 60 Pounds Gauge

(Wightman)

| Size Pipe | Equivalent Delivery in Cubic Feet of Compressed Air per Minute | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 9.84 | 14.73 | 19.64 | 24.60 | 29.45 | 34.44 | 39.35 | 49.20 | 58.90 | 68.6 | 78.6 | 88.4 | 98.4 | 118.1 | 137.5 | 156.6 | 176.5 | 196.4 | 294.5 | 393.7 | 492 | 589 | 686 | 786 | 884 | 984 |
| | Delivery in Cubic Feet of Free Air per Minute | | | | | | | | | | | | | | | | | | | | | | | | | |
| | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 250 | 300 | 350 | 400 | 450 | 500 | 600 | 700 | 800 | 900 | 1,000 | 1,500 | 2,000 | 2,500 | 3,000 | 3,500 | 4,000 | 4,500 | 5,000 |
| 1 in. | 18.24 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 1¼ " | 5.06 | 11.34 | 20.16 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 1½ " | 1.95 | 4.33 | 7.79 | 12.23 | 17.53 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 2 " | .42 | .95 | 1.69 | 2.65 | 3.80 | 5.17 | 6.77 | 10.61 | 15.20 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 2½ " | .13 | .29 | .52 | .81 | 1.16 | 1.58 | 2.09 | 3.24 | 4.65 | 6.31 | 8.28 | 10.47 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 3 " | .05 | .11 | .19 | .30 | .44 | .59 | .78 | 1.22 | 1.78 | 2.37 | 3.11 | 3.94 | 4.88 | 7.03 | 9.52 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 3½ " | .... | .05 | .08 | .13 | .19 | .26 | .36 | .55 | .78 | 1.07 | 1.40 | 1.77 | 2.20 | 3.17 | 4.29 | 5.57 | 7.08 | 8.77 | .... | .... | .... | .... | .... | .... | .... | .... |
| 4 " | .... | .... | .04 | .07 | .09 | .13 | .17 | .27 | .38 | .53 | .69 | .88 | 1.08 | 1.56 | 2.12 | 2.75 | 3.49 | 4.33 | 9.73 | .... | .... | .... | .... | .... | .... | .... |
| 4½ " | .... | .... | .... | .03 | .05 | .07 | .09 | .15 | .21 | .29 | .39 | .48 | .60 | .87 | 1.17 | 1.52 | 1.94 | 2.40 | 5.39 | 9.65 | .... | .... | .... | .... | .... | .... |
| 5 " | .... | .... | .... | .... | .03 | .04 | .06 | .08 | .12 | .17 | .22 | .28 | .34 | .49 | .67 | .87 | 1.17 | 1.37 | 3.08 | 5.51 | 8.61 | .... | .... | .... | .... | .... |
| 6 " | .... | .... | .... | .... | .... | .01 | .02 | .03 | .05 | .06 | .08 | .11 | .14 | .19 | .27 | .34 | .43 | .54 | 1.20 | 2.16 | 3.36 | 4.82 | 6.54 | .... | .... | .... |
| 7 " | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .03 | .04 | .05 | .06 | .09 | .12 | .15 | .19 | .24 | .55 | .98 | 1.53 | 2.19 | 2.97 | 3.91 | 4.94 | 6.19 |
| 8 " | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .01 | .02 | .03 | .04 | .06 | .08 | .09 | .12 | .27 | .41 | .77 | 1.11 | 1.50 | 1.98 | 2.51 | 3.10 |
| 9 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .03 | .04 | .05 | .06 | .15 | .27 | .42 | .61 | .83 | 1.08 | 1.36 | 1.69 |
| 10 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .02 | .03 | .03 | .04 | .09 | .16 | .25 | .36 | .48 | .63 | .79 | .99 |
| 12 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .03 | .06 | .09 | .14 | .19 | .25 | .32 | .39 |
| 14 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .03 | .04 | .06 | .09 | .11 | .15 | .18 |
| 16 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .02 | .03 | .04 | .05 | .07 | .09 |

**For longer or shorter pipes, the friction loss is proportional to the length—that is, for 500 feet, ½ of the above; for 4,000 feet, four times the above; etc.**

sponsible for this loss, the latter cannot be charged against the transmitting devices.

With the air cooled to normal temperature in the system, and assuming for the present that there is no leakage, there can be no loss of power during transmission. A drop in pressure there will be, in an amount determined by the physical characteristics of the transmission system. But since the conditions are isothermal—that is, $PV$ = a constant—this drop in pressure is accompanied with a corresponding increase in volume, and the intrinsic energy of the air in transit remains practically the same.

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(Wightman)

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|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 9.84 | 14.73 | 19.64 | 24.60 | 29.45 | 34.44 | 39.35 | 49.20 | 58.90 | 68.6 | 78.6 | 88.4 | 98.4 | 118.1 | 137.5 | 156.6 | 176.5 | 196.4 | 294.5 | 393.7 | 492 | 589 | 686 | 786 | 884 | 984 |
| | Delivery in Cubic Feet of Free Air per Minute | | | | | | | | | | | | | | | | | | | | | | | | | |
| | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 250 | 300 | 350 | 400 | 450 | 500 | 600 | 700 | 800 | 900 | 1,000 | 1,500 | 2,000 | 2,500 | 3,000 | 3,500 | 4,000 | 4,500 | 5,000 |
| 1 in. | 18.24 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 1¼ " | 5.06 | 11.34 | 20.16 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 1½ " | 1.95 | 4.33 | 7.79 | 12.23 | 17.53 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 2 " | .42 | .95 | 1.69 | 2.65 | 3.80 | 5.17 | 6.77 | 10.61 | 15.20 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 2½ " | .13 | .29 | .52 | .81 | 1.16 | 1.58 | 2.09 | 3.24 | 4.65 | 6.31 | 8.28 | 10.47 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 3 " | .05 | .11 | .19 | .30 | .44 | .59 | .78 | 1.22 | 1.78 | 2.37 | 3.11 | 3.94 | 4.88 | 7.03 | 9.52 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 3½ " | .... | .05 | .08 | .13 | .19 | .26 | .36 | .55 | .78 | 1.07 | 1.40 | 1.77 | 2.20 | 3.17 | 4.29 | 5.57 | 7.08 | 8.77 | .... | .... | .... | .... | .... | .... | .... | .... |
| 4 " | .... | .... | .04 | .07 | .09 | .13 | .17 | .27 | .38 | .53 | .69 | .88 | 1.08 | 1.56 | 2.12 | 2.75 | 3.49 | 4.33 | 9.73 | .... | .... | .... | .... | .... | .... | .... |
| 4½ " | .... | .... | .... | .03 | .05 | .07 | .09 | .15 | .21 | .29 | .39 | .48 | .60 | .87 | 1.17 | 1.52 | 1.94 | 2.40 | 5.39 | 9.65 | .... | .... | .... | .... | .... | .... |
| 5 " | .... | .... | .... | .... | .03 | .04 | .06 | .08 | .12 | .17 | .22 | .28 | .34 | .49 | .67 | .87 | 1.17 | 1.37 | 3.08 | 5.51 | 8.61 | .... | .... | .... | .... | .... |
| 6 " | .... | .... | .... | .... | .... | .01 | .02 | .03 | .05 | .06 | .08 | .11 | .14 | .19 | .27 | .34 | .43 | .54 | 1.20 | 2.16 | 3.36 | 4.82 | 6.54 | .... | .... | .... |
| 7 " | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .03 | .04 | .05 | .06 | .09 | .12 | .15 | .19 | .24 | .55 | .98 | 1.53 | 2.19 | 2.97 | 3.91 | 4.94 | 6.19 |
| 8 " | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .01 | .02 | .03 | .04 | .06 | .08 | .09 | .12 | .27 | .41 | .77 | 1.11 | 1.50 | 1.98 | 2.51 | 3.10 |
| 9 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .03 | .04 | .05 | .06 | .15 | .27 | .42 | .61 | .83 | 1.08 | 1.36 | 1.69 |
| 10 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .02 | .03 | .03 | .04 | .09 | .16 | .25 | .36 | .48 | .63 | .79 | .99 |
| 12 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .03 | .06 | .09 | .14 | .19 | .25 | .32 | .39 |
| 14 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .03 | .04 | .06 | .09 | .11 | .15 | .18 |
| 16 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .02 | .03 | .04 | .05 | .07 | .09 |

For longer or shorter pipes, the friction loss is proportional to the length—that is, for 500 feet, ½ of the above; for 4,000 feet, four times the above; etc.

# TABLE XII

## Loss of Pressure in Pounds, by Friction, in Transmission of Air through Pipes 1,000 Feet Long

Copyright, 1906, by Ingersoll-Rand Company

Initial Air-Pressure, 80 Pounds Gauge

(Wightman)

| SIZE PIPE | EQUIVALENT DELIVERY IN CUBIC FEET OF COMPRESSED AIR PER MINUTE | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 7.74 | 11.3 | 15.2 | 19.4 | 23.2 | 27.2 | 31.0 | 38.7 | 46.5 | 54.2 | 62.0 | 69.7 | 77.4 | 92.9 | 108.2 | 124.0 | 139.5 | 152 | 232 | 310 | 387 | 465 | 542 | 620 | 697 | 774 |
| | DELIVERY IN CUBIC FEET OF FREE AIR PER MINUTE | | | | | | | | | | | | | | | | | | | | | | | | | |
| | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 250 | 300 | 350 | 400 | 450 | 500 | 600 | 700 | 800 | 900 | 1,000 | 1,500 | 2,000 | 2,500 | 3,000 | 3,500 | 4,000 | 4,500 | 5,000 |
| 1 in. | 14.31 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 1¼ " | 3.96 | 8.46 | 15.31 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 1½ " | 1.53 | 3.26 | 5.92 | 9.64 | 13.79 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 2 " | .33 | .71 | 1.28 | 2.09 | 2.99 | 4.09 | 5.34 | 8.32 | 12.01 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 2½ " | .10 | .21 | .39 | .64 | .91 | 1.25 | 1.63 | 2.54 | 3.67 | 4.99 | 6.53 | 8.25 | 10.81 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 3 " | .03 | .08 | .14 | .24 | .34 | .47 | .61 | .96 | 1.38 | 1.88 | 2.45 | 3.13 | 3.83 | 5.61 | 7.46 | 9.86 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 3½ " | .01 | .03 | .06 | .11 | .15 | .21 | .27 | .43 | .62 | .84 | 1.11 | 1.40 | 1.73 | 2.46 | 3.37 | 4.42 | 5.61 | 6.64 | 15.41 | .... | .... | .... | .... | .... | .... | .... |
| 4 " | .... | .01 | .03 | .05 | .07 | .10 | .13 | .21 | .30 | .41 | .54 | .69 | .85 | 1.22 | 1.66 | 2.18 | 2.77 | 3.29 | 7.62 | 13.62 | .... | .... | .... | .... | .... | .... |
| 4½ " | .... | .... | .02 | .03 | .04 | .06 | .07 | .12 | .17 | .23 | .30 | .38 | .47 | .68 | .92 | 1.19 | 1.54 | 1.82 | 4.24 | 7.58 | 11.79 | .... | .... | .... | .... | .... |
| 5 " | .... | .... | .01 | .01 | .02 | .03 | .04 | .07 | .09 | .13 | .17 | .22 | .27 | .39 | .53 | .69 | .88 | 1.04 | 2.43 | 4.32 | 6.88 | 9.72 | 13.25 | .... | .... | .... |
| 6 " | .... | .... | .... | .... | .01 | .01 | .01 | .02 | .03 | .05 | .06 | .08 | .10 | .15 | .20 | .27 | .34 | .40 | .95 | 1.69 | 2.64 | 3.79 | 5.27 | 6.78 | 8.54 | 10.55 |
| 7 " | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .03 | .04 | .05 | .06 | .09 | .12 | .15 | .18 | .43 | .77 | 1.19 | 1.73 | 2.35 | 3.07 | 3.89 | 4.79 |
| 8 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .01 | .02 | .03 | .04 | .06 | .08 | .09 | .22 | .39 | .60 | .87 | 1.19 | 1.55 | 1.97 | 2.46 |
| 9 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .02 | .02 | .03 | .04 | .05 | .12 | .21 | .33 | .48 | .65 | .85 | 1.08 | 1.33 |
| 10 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .02 | .03 | .06 | .12 | .19 | .28 | .37 | .49 | .66 | .77 |
| 12 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .01 | .02 | .04 | .07 | .11 | .15 | .19 | .25 | .30 |
| 14 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .02 | .03 | .05 | .06 | .09 | .11 | .14 |
| 16 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .03 | .04 | .05 | .07 |

**For longer or shorter pipes, the friction loss is proportional to the length—that is, for 500 feet, ½ of the above; for 4,000 feet, four times the above; etc.**

## TABLE XIII

### Loss of Pressure in Pounds, by Friction, in Transmission of Air through Pipes 1,000 Feet Long

Copyright, 1906, by Ingersoll-Rand Company

Initial Air-Pressure, 100 Pounds Gauge

(Wightman)

| Size Pipe | Equivalent Delivery in Cubic Feet of Compressed Air per Minute | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 6.41 | 9.61 | 12.81 | 15.81 | 19.22 | 22.39 | 25.62 | 31.62 | 38.44 | 44.78 | 51.24 | 57.65 | 63.24 | 76.88 | 89.56 | 102.5 | 115.3 | 126.5 | 192.2 | 256.2 | 316.2 | 384.4 | 447.8 | 512.4 | 576.5 | 632.4 |
| | Delivery in Cubic Feet of Free Air per Minute | | | | | | | | | | | | | | | | | | | | | | | | | |
| | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 250 | 300 | 350 | 400 | 450 | 500 | 600 | 700 | 800 | 900 | 1,000 | 1,500 | 2,000 | 2,500 | 3,000 | 3,500 | 4,000 | 4,500 | 5,000 |
| 1 in. | 11.89 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 1¼ " | 3.29 | 7.42 | 13.20 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 1½ " | 1.28 | 2.87 | 5.11 | 7.75 | 11.42 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 2 " | .27 | .62 | 1.15 | 1.68 | 2.48 | 3.36 | 4.43 | 6.72 | 9.95 | 13.41 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 2½ " | .08 | .19 | .34 | .52 | .76 | 1.03 | 1.36 | 2.06 | 3.04 | 4.11 | 5.40 | 6.85 | 8.21 | 12.21 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 3 " | .03 | .07 | .12 | .19 | .29 | .39 | .51 | .77 | 1.14 | 1.54 | 2.06 | 2.57 | 3.08 | 4.58 | 6.19 | 8.13 | 10.23 | 12.39 | .... | .... | .... | .... | .... | .... | .... | .... |
| 3½ " | .01 | .03 | .05 | .08 | .13 | .17 | .23 | .35 | .51 | .69 | .92 | 1.16 | 1.39 | 2.14 | 2.79 | 3.67 | 4.64 | 5.60 | 12.81 | .... | .... | .... | .... | .... | .... | .... |
| 4 " | .... | .01 | .02 | .04 | .06 | .09 | .12 | .17 | .25 | .34 | .45 | .57 | .68 | 1.03 | 1.38 | 1.81 | 2.29 | 2.76 | 6.68 | 11.35 | .... | .... | .... | .... | .... | .... |
| 4½ " | .... | .... | .01 | .02 | .03 | .04 | .06 | .09 | .14 | .19 | .25 | .32 | .38 | .57 | .77 | 1.00 | 1.27 | 1.23 | 3.51 | 6.61 | 9.56 | 14 04 | .... | .... | .... | .... |
| 5 " | .... | .... | .... | .01 | .02 | .03 | .04 | .05 | .08 | .11 | .15 | .18 | .22 | .33 | .44 | .57 | .76 | .88 | 2.03 | 3.62 | 5.51 | 8.11 | 10.95 | 14.48 | .... | .... |
| 6 " | .... | .... | .... | .... | .01 | .01 | .02 | .02 | .03 | .04 | .05 | .07 | .08 | .12 | .17 | .22 | .28 | .34 | .78 | 1.41 | 2.14 | 3.16 | 4.26 | 5.59 | 7.04 | 8.51 |
| 7 " | .... | .... | .... | .... | .... | .... | .01 | .01 | .01 | .02 | .03 | .03 | .04 | .05 | .07 | .10 | .13 | .16 | .36 | .67 | .97 | 1.44 | 1.93 | 2.55 | 3.22 | 3.88 |
| 8 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .02 | .03 | .04 | .05 | .06 | .08 | .18 | .33 | .49 | .76 | .98 | 1.30 | 1.84 | 1.98 |
| 9 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .02 | .03 | .04 | .04 | .09 | .18 | .27 | .39 | .53 | .72 | .89 | 1.07 |
| 10 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .02 | .03 | .05 | .10 | .16 | .23 | .31 | .41 | .52 | .63 |
| 12 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .01 | .02 | .04 | .06 | .09 | .12 | .16 | .21 | .25 |
| 14 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | ... | .01 | .02 | .03 | .04 | .05 | .07 | .09 | .11 |
| 16 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .03 | .04 | .05 | .06 |

For longer or shorter pipes, the friction loss is proportional to the length—that is, for 500 feet, ½ of the above; for 4,000 feet, four times the above; etc.

# TABLE XIV

## Loss of Pressure in Pounds, by Friction, in Transmission of Air through Pipes 1,000 Feet Long

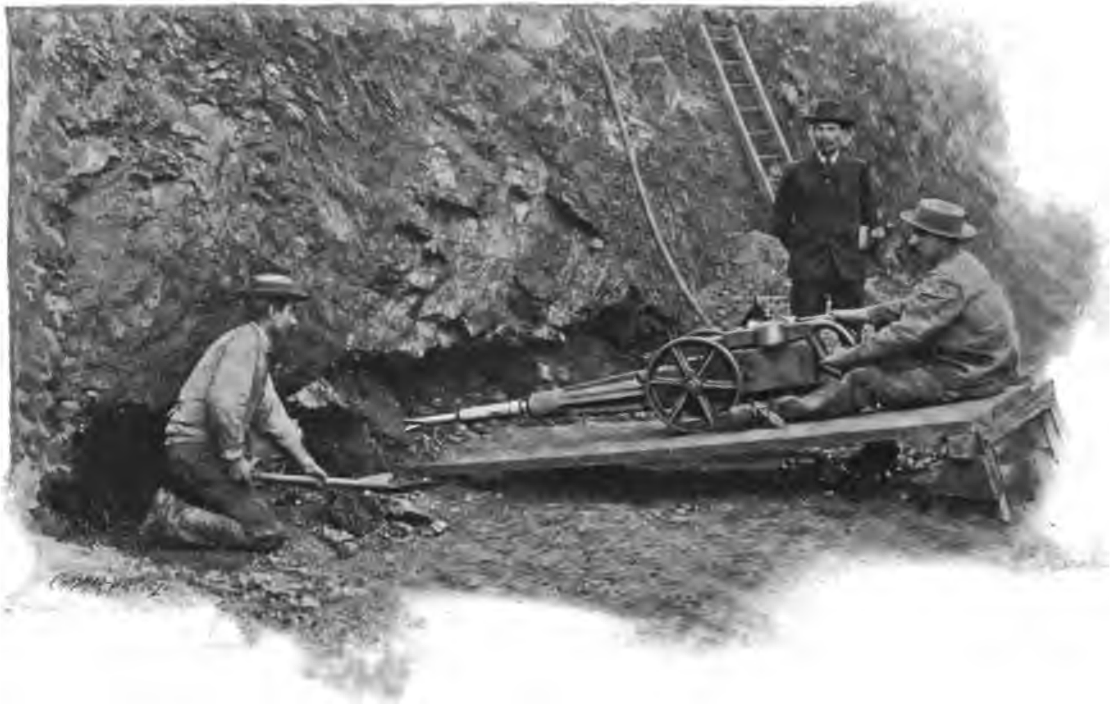
Copyright, 1906, by Ingersoll-Rand Company

Initial Air-Pressure, 125 Pounds Gauge

(E. F. Schaefer)

| Size Pipe | Equivalent Delivery in Cubic Feet of Compressed Air per Minute | | | | | | | | | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 5.26 | 7.89 | 10.51 | 13.15 | 15.79 | 18.41 | 21.05 | 26.30 | 31.58 | 36.81 | 42.10 | 47.30 | 52.60 | 63.20 | 73.70 | 84.20 | 94.70 | 105.1 | 157.9 | 210.5 | 263.0 | 315.8 | 368.1 | 422.0 | 473.0 | 526.0 |
| | Delivery in Cubic Feet of Free Air per Minute | | | | | | | | | | | | | | | | | | | | | | | | | |
| | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 250 | 300 | 350 | 400 | 450 | 500 | 600 | 700 | 800 | 900 | 1,000 | 1,500 | 2,000 | 2,500 | 3,000 | 3,500 | 4,000 | 4,500 | 5,000 |
| 1 in. | 9.88 | 22.20 | 39.50 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 1¼ " | 2.70 | 6.07 | 10.82 | 16.88 | 24.33 | 33.05 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 1½ " | 1.05 | 2.37 | 4.22 | 6.58 | 9.47 | 12.90 | 16.84 | 26.30 | 37.90 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 2 " | .23 | .51 | .91 | 1.42 | 2.04 | 2.78 | 3.63 | 5.68 | 8.18 | 11.08 | 14.51 | 18.38 | 22.68 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 2½ " | .07 | .16 | .28 | .43 | .63 | .85 | 1.11 | 1.73 | 2.51 | 3.39 | 4.44 | 5.61 | 6.95 | 10.00 | 13.60 | 17.80 | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... |
| 3 " | .03 | .06 | .10 | .16 | .23 | .32 | .42 | .65 | .94 | 1.27 | 1.67 | 2.11 | 2.61 | 3.76 | 5.11 | 6.68 | 8.45 | 10.42 | 23.48 | .... | .... | .... | .... | .... | .... | .... |
| 3½ " | .01 | .03 | .05 | .07 | .11 | .14 | .19 | .29 | .42 | .58 | .75 | .95 | 1.18 | 1.69 | 2.31 | 3.01 | 3.81 | 4.71 | 10.59 | 18.81 | 29.40 | .... | .... | .... | .... | .... |
| 4 " | .... | .01 | .02 | .04 | .05 | .07 | .09 | .15 | .21 | .28 | .37 | .47 | .58 | .84 | 1.14 | 1.49 | 1.88 | 2.32 | 5.23 | 9.30 | 14.52 | 20.90 | 28.51 | .... | .... | .... |
| 4½ " | .... | .... | .01 | .02 | .03 | .04 | .05 | .08 | .12 | .16 | .21 | .26 | .32 | .46 | .63 | .83 | 1.04 | 1.29 | 2.90 | 5.15 | 8.05 | 11.59 | 15.78 | 20.61 | 26.10 | 32.20 |
| 5 " | .... | .... | .... | .01 | .02 | .02 | .03 | .05 | .07 | .09 | .12 | .15 | .18 | .27 | .36 | .47 | .60 | .74 | 1.65 | 2.94 | 4.60 | 6.63 | 9.01 | 11.80 | 14.90 | 18.45 |
| 6 " | .... | .... | .... | .... | .01 | 01 | .01 | .02 | .03 | .04 | .05 | .06 | .07 | .10 | .14 | .18 | .23 | .29 | .64 | 1.15 | 1.80 | 2.59 | 3.53 | 4.61 | 5.83 | 7.20 |
| 7 " | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .02 | .03 | .03 | .05 | .06 | .08 | .11 | .13 | .29 | .52 | .82 | 1.18 | 1.61 | 2.19 | 2.65 | 3.27 |
| 8 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .01 | .02 | .02 | .03 | .04 | .05 | .07 | .15 | .26 | .41 | .60 | .81 | 1.06 | 1.34 | 1.65 |
| 9 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .01 | .02 | .02 | .03 | .04 | .08 | .15 | .23 | .33 | .45 | .58 | .73 | .90 |
| 10 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .01 | .02 | .02 | .05 | .08 | .13 | .19 | .26 | .34 | .43 | .53 |
| 12 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .01 | .02 | .03 | .05 | .07 | .10 | .13 | .17 | .21 |
| 14 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .02 | .02 | .0[illegible] | .04 | .06 | .08 | .10 |
| 16 " | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .... | .01 | .01 | .02 | .02 | .03 | .04 | .05 |

For longer or shorter pipes, the friction loss is proportional to the length—that is, for 500 feet, ½ of the above; for 4,000 feet, four times the above; etc.



SULLIVAN PNEUMATIC PICK MACHINE AT WORK IN OPEN COAL PIT

### TABLE XV

### Globe Valves, Tees, and Elbows

The reduction of pressure produced by globe valves is the same as that caused by the following additional lengths of straight pipe, as calculated by the formula,

$$\text{Additional length of pipe} = \frac{114 \times \text{Diameter of pipe}}{1 + (3.6 \div \text{Diameter})}:$$

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Diameter of pipe | 1 | 1½ | 2 | 2½ | 3 | 3½ | 4 | 5 | 6 inches |
| Additional length | 2 | 4 | 7 | 10 | 13 | 16 | 20 | 28 | 36 feet |
| Diameter of pipe | 7 | 8 | 10 | 12 | 15 | 18 | 20 | 22 | 24 inches |
| Additional length | 44 | 53 | 70 | 88 | 115 | 143 | 162 | 181 | 200 feet |

The reduction of pressure produced by elbows and tees is equal to two-thirds of that caused by globe valves. The following are the additional lengths of straight pipe to be taken into account for elbows and tees. For globe valves, multiply by $\frac{3}{2}$:

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Diameter of pipe | 1 | 1½ | 2 | 2½ | 3 | 3½ | 4 | 5 | 6 inches |
| Additional length | 2 | 3 | 5 | 7 | 9 | 11 | 13 | 19 | 24 feet |
| Diameter of pipe | 7 | 8 | 10 | 12 | 15 | 18 | 20 | 22 | 24 inches |
| Additional length | 30 | 35 | 47 | 59 | 77 | 96 | 108 | 120 | 134 feet |

These additional lengths of pipe for globe valves, elbows, and tees must be added in each case to the actual length of straight pipe. Thus a 6-inch pipe 500 feet long, with 1 globe valve, 2 elbows, and 3 tees, would be equivalent to a straight pipe 500 + 36 + (2 × 24) + (3 × 24) = 656 feet long.

60, 80, 100, and 125 lbs.—for volumes of from 50 to 5,000 cubic feet of free air per minute, and for a distance of 1,000 feet. For longer or shorter distances, the drop in pressure will be practically proportional—that is, one-half as much for 500 feet, and four times as much for 4,000 feet. The delivered volume of compressed air given in these tables is the equivalent volume at initial pressure, and is therefore not strictly correct at the point of delivery, since the pressure will be decreased by the amount given, and the volume increased accordingly. However, since these are minimum volumes, they may be safely used where the reduced pressure is taken into account. These tables show that it is possible to get almost any pressure drop wanted, and that this loss of pressure can be practically eliminated if a pipe large enough is used. But here the question of cost of pipe and fittings enters in, and decision in any case will be a compromise for a moderate drop of pressure in connection with a moderate cost of line.

**Effect of Valves, Tees, and Elbows.** Supplementing the four tables just referred to, is Table XV, also furnished by the Ingersoll-Rand Company, giving the effect of valves, tees, and elbows in a pipe-line, expressed in terms of additional lengths of pipe. The data given in these five tables will furnish a solution for almost any problem of transmission which will be encountered in average practice.

**Necessity for Dry Air.** There are many reasons why air should be as dry—that is, as free from water vapor—as possible, during transmission. At the outset, let it be clearly understood that air cannot "freeze up." It is the water in the air which freezes when the conditions are favorable.

The capacity of air for carrying water vapor increases with its temperature. This means that the cooler the air, the less moisture it will contain; and as temperature falls, the water vapor is condensed to water, and accumulates wherever opportunity offers. If this water is in the pipe-lines, it will freeze up in cold weather, clogging and ultimately choking the line, and frequently bursting the pipe. Or, the small particles of ice may be carried along with the movement of the air until they encounter a valve or other obstruction, where they will gather and eventually fill the pipe, closing the air-passage. Even in warm weather, the accumulation of water in the pipe results in *water-hammer* and leaky joints. It flows to the low points in the line, and fills the pipe, when it must be pushed ahead by the air-pressure, resulting in an added loss of pressure.

Water in the air is the cause of freezing at exhaust ports of drills, pumps, etc., since the sudden expansion of the air on exhaust produces such a low temperature that ice is formed and the exhaust is clogged, oftentimes even in warm weather. There can be no difficulty of this kind where the air is dry.

Water carried with the air into the cylinders of drills, pumps, pneumatic tools, and other compressed-air appliances, condenses on the walls and excludes the lubricant, resulting in an added friction load and more rapid wear and leakage.

All of these points emphasize the importance of removing the water from the air, so far as this is practically possible. The logical sequence of operations in which this should be accomplished, is: *first,* before compression; *second,* during compression; *third,* after compression and before transmission; *fourth,* during transmission. In all of these cases the problem is one of adequate cooling.

**The Antecooler.** Although not properly a part of the transmission system, the antecooler may here be considered in its relation to the question of dry air. It is not generally used, and has not received the attention its importance deserves. It is very similar in construction to the intercooler of a compound compressor, and is placed on the air intake to the compressor, so that all air entering the compressor must pass through it. Of course it must be larger in proportion, to provide for the larger volume and lower velocity of the atmospheric air passing through it, which must not be perceptibly rarified in passing. If the intake air is warm and moist, and water sufficiently cool can be had for the antecooler, this device will condense a large part of the moisture before it enters the compressor. Aside from the drying effect, the antecooler may have an important effect upon the economy of the air-power system, when it is remembered that for every five-degree reduction in the temperature of the intake air there is an increase of about one per cent in volumetric efficiency. As to the design of the antecooler, it need only be said that it should conform to the requirements already outlined in connection with the intercooler.

**Removal of Moisture during Compression.** The removal of the moisture from the air during the compression process has already been considered in the discussion of Compound Compression, in which it was seen that the intercooler played an important part in this process by condensing the water vapor between the stages of compression. No further reference need here be made to this point.

**The Aftercooler.** The aftercooler is practically identical with the intercooler, it being important to remember, however, that it must deal with higher pressures and smaller volumes. It should be placed as close as possible to the air-compressor, and provided with a suitable means of draining it. The function of the aftercooler is to reduce the temperature of the compressed air as low as possible, condensing as much as possible of the moisture and delivering the air to the line at maximum density and minimum temperature.

Aside from the withdrawal of moisture, the aftercooler has a most important effect on pipe-line efficiency. Nothing about the air-power plant will waste more power than a badly leaking pipe-line; and nothing is so conducive to a leaky line as wide extremes of temperature, producing large strains of expansion and contraction. Without an aftercooler, air enters the pipe-line hot, while the pipe itself is cold; but the latter gradually takes up heat from the air, and the heated portion extends along the line the longer the load is on the line; and the whole system expands accordingly. When the plant is shut down—probably at night, when the atmosphere is cooler—there is a corresponding contraction all along the line. This goes on day after day, with alternate expansions and contractions; and the resultant difficulty in keeping tight joints and valves and in preventing leakage becomes evident, even where the best mechanical precautions are taken along the line. The use of an aftercooler obviates these extremes of temperature, and leaves the pipe-line subject only to the expansion and contraction consequent upon changes in atmospheric temperature.

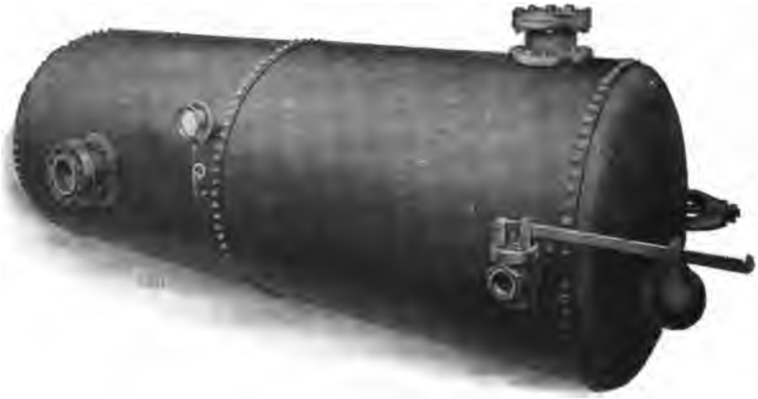


Fig. 78. Horizontal Air-Receiver.
Ingersoll-Rand Co., New York City.

**Primary Air-Receivers.** The discharge from the compressor is more or less pulsating in character, corresponding to the strokes of compression; and an air-receiver is a *rectifier*, so to speak, which receives and absorbs the pulsations and delivers a steady flow of air to the line. It is also in a very small degree an *accumulator*, in which excess energy is momentarily stored and momentarily withdrawn; but it cannot be relied upon as a power storage in this respect.

The air-receiver cannot be too large, nor can there be too many of them, provided that leakage is properly guarded against. Ample receiver capacity is especially useful on work of intermittent character, such as running rock-drills and pneumatic tools. It makes the problem of regulation simpler, the work of the governor or unloader easier, and helps out on overloads. In a broad sense, the air-receiver is the *balance wheel* of the air-power system, equalizing supply and demand and smoothing out the minor fluctuations in the load curve.

Preference should be given to the *vertical* receiver as against the *horizontal.* It should be placed as close as possible to the compressor or aftercooler, and a pipe amply large in section should be used for connecting it up. It is well to have the pipe entering the receiver from the compressor a size larger than the discharge pipe from the receiver to the line. Elbows should be avoided as far as possible in the former section of piping, and preference given to wide-sweep bends. No valve should be placed between the compressor and receiver unless it is protected by a relief-valve on the side nearest the compressor. It is perhaps unnecessary to state that each receiver should have a relief- or safety-valve; and where the receiver is out of doors, its safety-valve should be piped back into a warm place, to prevent its freezing up. The piping for the main or primary receiver should enter near the top, and leave near the bottom.

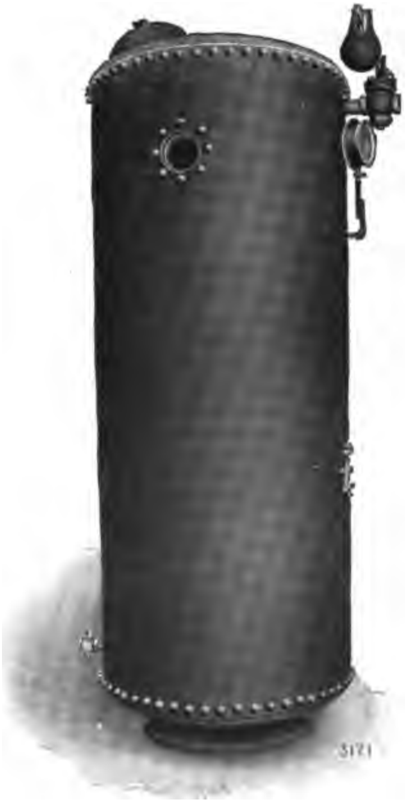


Fig. 79. Vertical Air-Receiver.
Ingersoll-Rand Co., New York City.

There is, of course, some cooling of the air in its passage through the receiver; and on this account several small receivers are better than one large one, as offering a larger cooling surface. Each receiver should be provided with a drain-cock at its lowest point, and this cock should be opened regularly and frequently for the withdrawal of water and oil. The best location for a receiver is out of doors, where it is exposed to the coolest atmosphere. A pressure-gauge on each receiver is convenient.

**Secondary Receivers.** Secondary receivers are essentially moisture traps, and, in large transmission systems, should be placed at every low point in the line, and always *below* the line, so that they may collect the water which condenses in the piping and flows to this point. They need not be very large, and are preferably of horizontal pattern; and the pipe-lines should enter and leave from the top or upper side, so that the flow of the air will not be impeded by the accumulated water. Each receiver should have a drain-cock at its lowest point, and should be drained regularly and often. Where only one secondary receiver is used, it should be placed far enough away from the primary receiver, so that the air may have a chance to cool to atmospheric temperature before entering it. All branches from the main transmission line should be taken off *beyond* the first secondary receiver. It is not necessary that these small tanks should have a safety-valve.

**The Pipe-Line.** The size of pipe to be used having been determined from the tables already given, or otherwise, the problem of transmission efficiency becomes one of care in the smaller details of installation. The most important consideration is that the pipe-line *shall not leak*. A transmission system which will not hold gauge pressure over night with all outlets and valves closed, cannot be considered a first-class system. More power can be wasted through a few seemingly insignificant leaks in the line than can be saved by the most refined methods of air-compression.

It is better to lay pipe on the surface of the ground than underground, for, in the former case, not only is there more cooling, and therefore more complete removal of moisture, but a leak is more readily discovered and stopped. If it is not desirable to lay the line on the surface, it should be laid in a trough or box so that it can be readily examined. A good way of testing a line for leaks is to shut down all the air outlets provided with valves, and observe how many strokes of the compressor are necessary to maintain gauge pressure. This will give a basis for figuring piston displacement which is necessary to carry the *leakage load*; and the results are frequently surprising. A pipe-line should be regularly inspected for leaks, by going over it with a brush and a pail of soapy water, "painting" each joint, branch, valve, or connection on pipe-line, receivers, etc., with this solution. A leak will be revealed by the soap bubbles formed, even when it is not sufficient to make a sound. Flanged joints are to be preferred to screwed joints, as more easily packed and kept tight. In making up a pipe-line, care should be exercised to see that none of the packing in the joints projects into the pipe, offering an obstruction; and each length of pipe should be cleaned of all obstructions before connecting up. Suitable supports should be provided at safe intervals, so that the line will not sag, with danger of sprung joints. Expansion and contraction must be provided for, and consequent leakage guarded against, either by the use of packed slip joints, or, preferably and more cheaply, by offsetting the line at intervals of a few hundred feet with wide-angle bends. Wide-sweep bends should be preferred to short elbows. Branches should be taken off with angle fittings pointing in the direction of flow, rather than with tees. Valves should be of gate pattern, rather than of globe type, as offering more direct passage and less friction. Rising-stem valves have the advantage over others, of showing at a glance whether the valve is closed or open, and the degree of opening. Every low place in the line where water might accumulate should have a secondary receiver, or at least a drain-cock, which should be opened frequently and the water blown off. Mr. Frank Richards is authority for the statement that "the velocity of air in main transmission lines should never exceed twenty feet per second; and in small branch pipes it should be still lower."

**The Loop System.** The usual air-transmission system consists of a main or trunk line, with branches and sub-branches taken off at intervals. Evidently a break at any point in the main line means the shutting down of all the machines connected up beyond the break, while repairs are in progress. To avoid this shut-down of a part of the air-transmission scheme, the *loop system* of distribution was devised. It is illustrated in the diagram, Fig 80. The main supply-pipe from the primary receiver is divided into two branches which later unite in a loop or closed circuit. From this main loop, other secondary loops are formed by intersecting and uniting branch pipes. Valves are placed as indicated by the cross marks in the diagram.

A little study of this sketch will show that nothing short of a break in the main between the receiver and the first branch forming the loop can shut down this system or interfere with its workings. Any section in any loop or sub-loop can be cut out for alterations or repairs, without in any way affecting the other sections. While any one branch of the main or secondary loops might be overloaded while supplying the whole load on that loop, yet the shut-down of one side of the loop would be only temporary, and the work could go on at slightly less efficiency in the transmission system while changes or repairs were in progress.

The loop system is probably not justified in any but comparatively large installations, such as air-power supply for shops, quarries, mines, and contracts where a large number of compressed-air appliances are in use continuously, and where it is important that the work shall not suffer interruption. This system will probably cost more to install than a straight trunk-and-branch system; but its extra cost may be looked upon as an insurance against total shut-down during repairs or alterations in the mains.

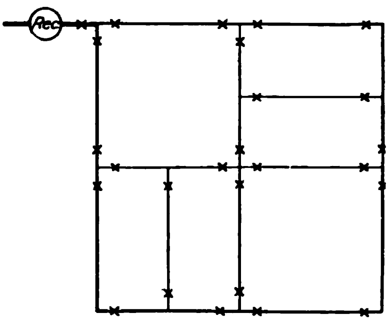


Fig. 80. The "Loop System" of Air Power Distribution.

**The Dense-Air System.** The usual practice in ordinary air-compression and application is to compress from atmosphere to a given pressure, transmit at this pressure, apply at this pressure, and exhaust to atmosphere. In the *dense-air* or *return-pipe system*, on the contrary, the exhaust from the air-motor is above atmosphere, and is piped back to the compressor intake, the cycle of compression, transmission, and expansion thus occurring in a closed circuit, and pressures always being well above atmosphere.

The dense-air system is based upon the fact that the power required in the first stages of compression, and for comparatively low pressures, is greater than that required in the higher stages of compression. For instance, it takes the same power to compress a given volume of free air from atmosphere to 30 lbs. gauge as to continue to compress it from 30 lbs. gauge to 90 lbs. gauge. In the first case, if the air at 30 lbs. is used in a motor discharging to atmosphere, the maximum effective pressure is 30 lbs. In the second case, if the air at 90 lbs. is used in a motor exhausting at 30 lbs., and this exhaust is piped back to the compressor intake for re-compression to 90 lbs., there will be a maximum effective pressure of 90–30, or 60 lbs., at the motor. The cost of compression in the latter case will be the same as in the first, but the maximum effective pressure is twice as great.

The higher the compression is carried, the greater the comparative advantage of the dense-air system. For example, the power required to compress from atmosphere to 60 lbs. gauge is practically the same as that needed to compress from 60 lbs. to 350 lbs. gauge. The maximum effective pressures are in these two cases 60 lbs. and 290 lbs., respectively, assuming that the dense-air system is used with the higher pressure. But the ordinary mechanical devices on the market are not adapted to more than 125 or 150 lbs. pressure, and this fixes the practical limit for the use of the dense-air system. It is to be remembered that even though the effective difference of pressures be only 125 lbs., the air supply line from compressor to motor must carry the full highest pressure of the system. This is another fact setting a limit to the maximum pressures to be employed in this system, as standard pipe and fittings are always to be preferred.

Aside from the saving in compressor power, the dense-air system also eliminates all losses due to clearance, since the air is in a closed circuit.

The complete dense-air system consists of a compressor (usually single-stage, since the ratio of intake and discharge pressures is low, even though these pressures themselves may be high), two pipe-lines, and a motor using the compressed air. There will be some leakage in operation, so that a small *booster* compressor is ordinarily used also, supplying air to the low-pressure side of the system to make good these leakage losses and maintain the proper ratio of pressures.

It must be confessed that the dense-air system does not fully realize all the economies which it promises; but the actual working-out of the system defining these shortcomings demands a mathematical investigation which is beyond the scope of this book. It is evident that the system is not adapted to intermittent service or widely fluctuating loads. Its best field of application is in the operation of direct-acting pumps by compressed air.

The general scheme of compression and expansion in a closed circuit, however, is used in three very practical devices on the market —namely, the *return-air pumping system*, the *electric-air rock-drill*, and the *electric-air channeler*, reference to which will be made later.

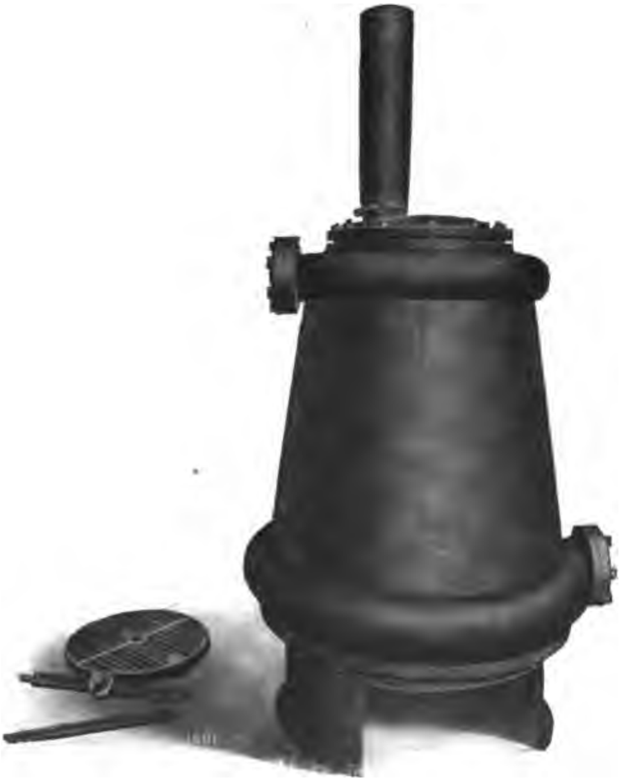


Fig. 81. Sergeant Air-Reheater.
Ingersoll-Rand Co., New York City.

**Reheating Compressed Air.** The reheating of compressed air might not seem properly to belong in a section on air-transmission; but it is here introduced because the definition of the transmission system here adopted covers all devices for delivering the compressed air from the compressor to the machine using it.

Reheating is, in brief, a process of restoring to the compressed air the heat which was withdrawn from it by the cooling devices during compression and transmission. If it were economically desirable and practically possible to retain all the heat generated in the air during compression and up to the point of application, reheating would be unnecessary; but it has been seen that economical compression and transmission demand the removal of all the heat possible.

It will be remembered that, according to Charles's law, the volume of air at a constant pressure varies directly as its absolute temperature. The reheating of compressed air at a constant pressure, therefore, produces an increase in volume at that pressure proportional to the degree of heating. In that respect it is equivalent to an increase in compressor capacity; but a vital difference is that this increased volume by reheating is secured much more cheaply than by increased compressor capacity. Mr. Frank Richards, in his book on "Compressed Air," has explained this very clearly; and the comparison which follows is used here with his permission, the figures all being theoretical:

> "Weight of 1 cu. ft. of steam at 75 lbs. gauge, 0.2089 lb.
>
> "Total heat units in 1 lb. of steam at 75 lbs. gauge, from water at 60° F., 1,151.
>
> "Total heat units in 1 cu. ft. of steam at 75 lbs. gauge, 1,151 × 0.2089 = 240.44.
>
> "To produce by compression with a steam-driven compressor 1 cu. ft. of compressed air at 75 lbs. gauge and 60° F., about 2 cu. ft. of steam at the same pressure are required; or the heat units employed in producing 1 cu. ft. of compressed air (at this pressure) will be about 240.44 × 2, or 480.88 heat units, which is the thermal cost of 1 cu. ft. of compressed air at the above temperature and pressure. . . . . The difference in the thermal cost of any volume of compressed air thus produced by mechanical compression and the cost of any additional volume that may result from the subsequent reheating of the air, is very striking.
>
> "Weight of 1 cu. ft. of free air at 60° F., 0.076 lb.
>
> "Weight of 1 cu. ft. of compressed air at 75 lbs. gauge and 60° F., 0.456 lb.
>
> "Units of heat required to double the volume of 1 lb. of air at 60° F., 123.84.
>
> "Units of heat required to double the volume of 1 cu. ft. of compressed air at 75 lbs. and 60° F., 123.84 × 0.456 = 56.47.
>
> "Cost of 1 cu. ft. of reheated compressed air at 75 lbs., compared with the cost of 1 cu. ft. as produced by ordinary compression in a steam-driven compressor, 480.88 to 56.4771, or 1 to 0.1174.
>
> "Here we see that the cost in heat units of the volume of air produced by reheating is less than one-eighth of the cost of the same volume produced by compression."

To double the volume by reheating, it will be necessary to double the absolute temperature. In the case just cited, the initial absolute temperature is 60 + 461, or 521° absolute; and doubling this gives 1,042° absolute, or 1,042 − 461 = 581° F. But air loses its heat rapidly, and it is probable that to secure a double volume *in the motor*, the air would have to be heated to at least 800° F. This temperature cannot be practically handled, and probably the best that should be attempted in practice is to increase the volume by reheating 50 per cent. In this latter case, assuming initial temperature of 521° absolute, the final temperature theoretically necessary to add 50 per cent to the volume will be 1½ × 521, or 728° absolute, corresponding to 321° F. A safe allowance for loss by cooling between a reheater and motor, would bring this up to 450° F.—an increase of 450 − 60, or 390° F. It was seen that 56.47 heat units were used in raising one cubic foot of compressed air at 75 lbs. from 60° to 581° F., the temperature increase being 521°. The heat units needed to reach 390 F. are found by the proportion 521 : 390 = 56.4 : 42.27.

In other words, the first cubic foot of compressed air at 75 lbs. required 480.88 heat units, and the additional half-foot secured by reheating required 42.27 heat units, making the total thermal cost of 1½ cubic feet of compressed air at 75 lbs. gauge 523.15 heat units, or at the rate of 348.78 heat units per cubic foot. The relative cost per cubic foot of compressed air at 75 lbs. gauge by compression alone, and by compression and reheating, is thus 480.88 : 348.78 = 1 : 0.72.

From the above it appears that the gain by reheating compressed air at 75 lbs. gauge to increase its volume one-half, is 28 per cent. Stating the relation otherwise, 0.72 : 1 = 1 : 1.38. In other words, the total fuel used in a system using compression and heating will give 38 per cent more work than can be had from the same amount of fuel by compression alone, without reheating.

There are reheaters of many different types on the market, and some which are not generally available. The more common ones may be divided into two classes: (1) Internal-combustion reheaters, in which the compressed air passes directly over the flame from burning oil or coke, the products of combustion going through into the motor; (2) reheaters in which the air is heated by passing through closed tubes or shells above or surrounding a fire of coke or coal. The former method is the more efficient, but it contaminates the air. The latter delivers the air clean and free from ash, soot, and foreign substances which may collect in the motor. Other types not in general use employ an electric heating coil in the air passage; or the air is heated by hot water; or steam coils furnish the necessary heat.

As to the construction of reheaters, the first essential is that they shall be made of a material which will not burn out, for it is to be remembered that they are not protected by water as are the tubes and plates of steam boilers. Cast iron seems to give the best results in this respect. Another requirement is that they shall give an ample heating surface without materially interfering with the free flow of air. A third essential is that leakage shall not develop under the expansion and contraction strains.

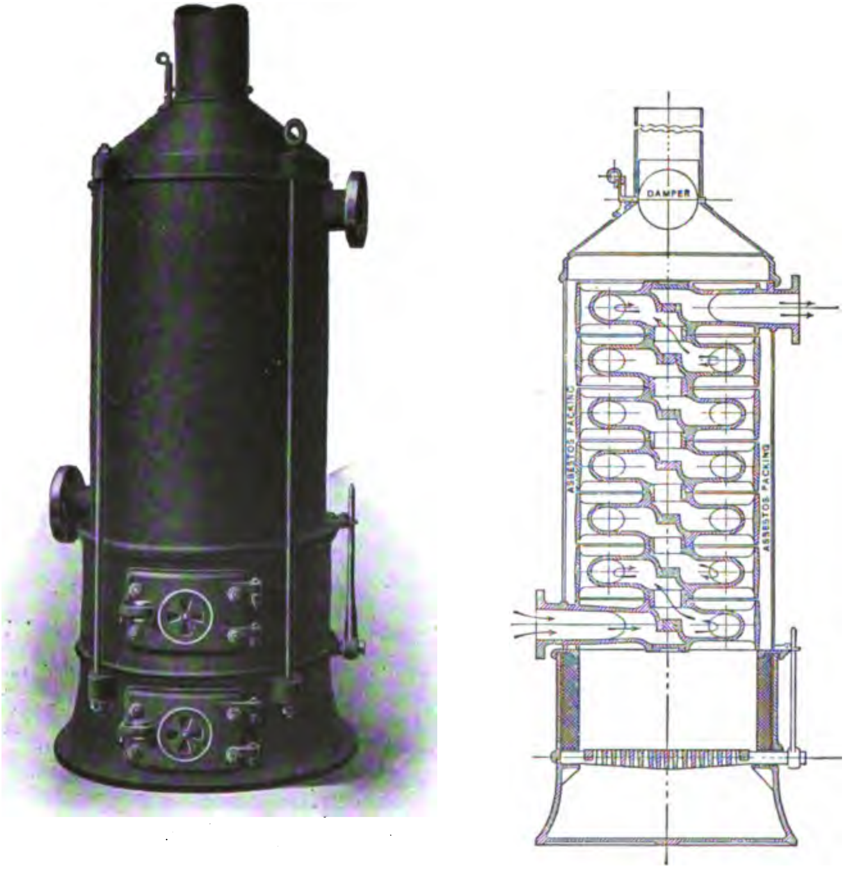


**Fig. 82. General and Sectional Views of Stationary Air-Reheater. Sullivan Machinery Co., Chicago, Ill.**

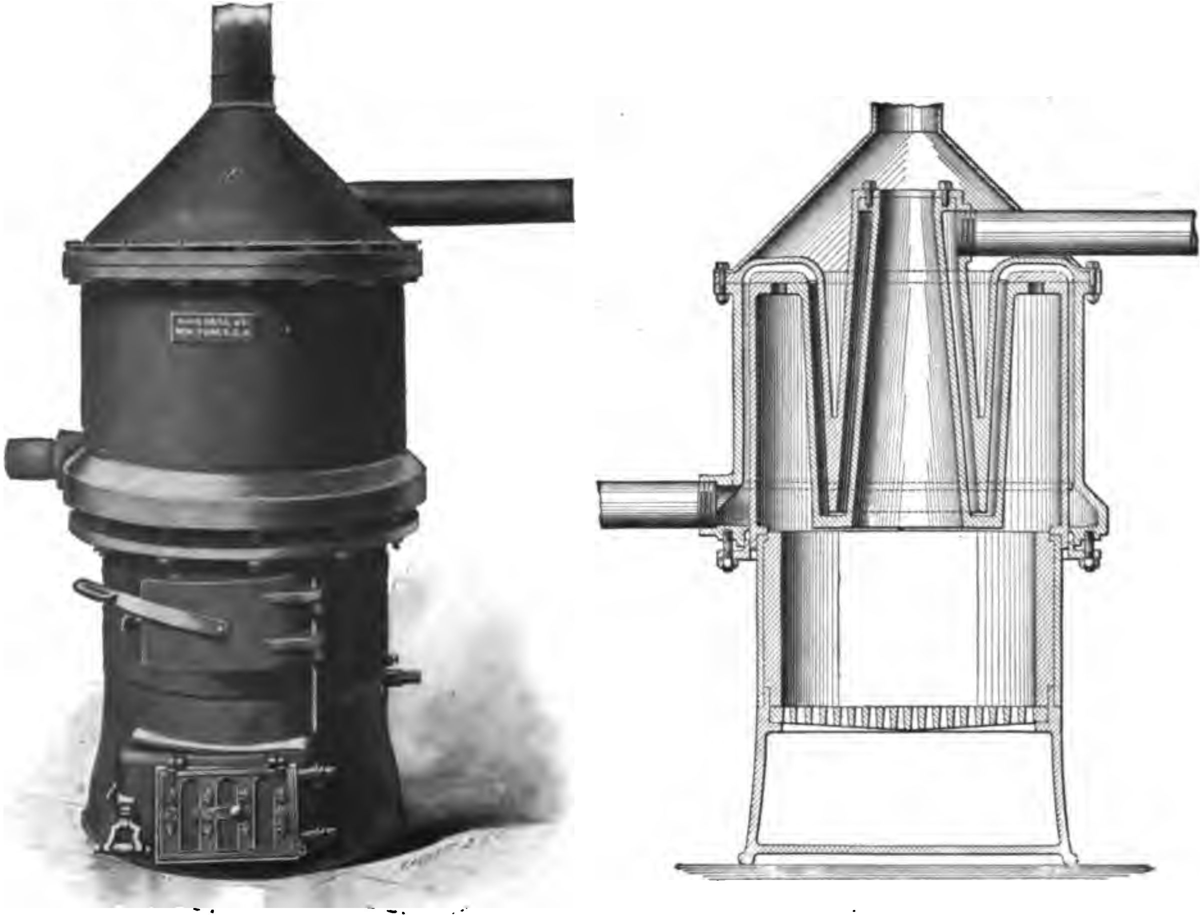


Fig. 83. Rand Air-Reheater. Sectional view shown at right. Ingersoll-Rand Co., New York City.

The reheater should be placed as close as possible to the work it supplies, and the connecting pipes should be well protected against loss of heat by radiation. Where the reheated air is used in a motor without expansion, particular care must be exercised in the matter of lubrication at this high temperature. Where the air is used expansively, the mean cylinder temperature in the motor will be lower and safer.

It is probable that reheating seldom pays, except where the air is used in large volumes and for comparatively steady service, such as in pumps, hoists, stone channelers, and air-motors of large power.

## APPLICATION OF COMPRESSED AIR

**Elementary Theory.** The phenomena of the expansion of compressed air are governed by the same laws as those holding for the compression of air. It was seen in the beginning of this treatise, that the compression of air produces an increase in temperature due to intermolecular action. Conversely, by a reversal of this molecular activity, the expansion of air produces a drop in temperature.

*Isothermal expansion* is expansion at constant temperature, according to Boyle's law, $PV$ = a constant. It is impossible of attainment, as is isothermal compression; but it is the ideal to be sought in the practical working of expansion air engines or motors.

*Adiabatic expansion* is expansion with a total proportionate drop in temperature, according to Charles's law governing the relation of pressures, volumes, and temperatures. Practical work approaches this, though it is the process to be avoided as far as possible. It is never fully realized, because there is always some absorption of heat from the cylinder in which expansion occurs.

**Air-Motor Indicator Cards.** In Fig. 84, the theoretical phenomena of compression and expansion are shown in an ideal indicator card. A volume of air compressed adiabatically will follow the curve $AB$, and, after cooling to initial temperature, will occupy the volume represented by $EC$. If it had been possible to compress it isothermally, the line of compression would have been the curve $AE$, and a considerable saving of power would have resulted, as represented by the area $ABE$, as already explained. If this volume of air could now be expanded isothermally to initial pressure, the expansion line would retrace the curve $AE$; and the original volume, at original temperature and pressure, would be recovered, with a restoration of all the work expended in isothermal compression and represented by the area *AECD*. If it should expand adiabatically, however, it would follow the curve *EF* with a loss of work, as compared to isothermal, represented by the area *EAF*, the volume at initial pressure and at the lower temperature represented by the line *AD* showing a decrease in volume represented by *AF*. The total loss, therefore, in adiabatic compression and expansion (assuming a perfect intermediate cooling to initial temperature), is represented by the area *ABEF*.

As a matter of fact, however, compression will more nearly follow the line *AJ*, on account of the cooling devices used on the compressor; and expansion will possibly follow the line *EK*, owing to the absorption of heat from the motor. The loss of work, therefore, between practical compression and practical expansion, is probably more correctly represented by the area *AJEK*, so that the conditions are not quite so bad as they might at first appear.

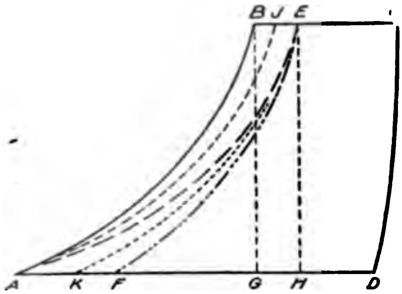


**Fig. 84.** Curves of Air Compression and Expansion.

The operation thus far discussed has assumed that the air after compression was allowed to cool to its initial temperature before it was expanded. If it were possible, however, to compress adiabatically, and then, without any loss of the heat of compression, to expand adiabatically, it is evident that both compression and expansion would follow the curve *AB*; and while more work was required in compression, still the full equivalent of this would be returned in expansion. This is plainly impossible and impracticable.

But assume that compression follows the practical curve *AJ*; that the air is cooled to initial temperature and transmitted, its volume then being represented by *EC*; that enough heat is applied to this transmitted volume to re-expand it to the volume *JC*; and that at this temperature and volume the air is allowed to expand along the practical curve *JA*. This is a cycle of operations quite possible of attainment, and represents good, everyday compression, cooling, and transmission, reheating to increase the volume, and expansion along practical lines. Reheating of the air to increase its volume from *EC* to *JC* has saved the power (as compared with cool air) shown by the area *AJEK*. In the discussion of reheating in a previous section, it was shown that this gain is secured at a relatively small cost.

It was seen, in discussing air-compression, that the ideal method of cooling would be that in which the heat was removed as fast as produced, compression following the isothermal curve; but since this is impossible in practice, the next best arrangement is the division of compression into stages, with cooling to original temperature between each compression.

Similarly, the ideal expansion air-motor would be one in which the isothermal curve was followed by supplying, during expansion, the heat necessary to maintain a constant temperature. This being impossible in practice, the nearest approach to the process would be one of stage expansion and interheating, which has been proposed, and which, indeed, has been used to a limited extent. But the problem of supplying heat is more difficult than that of withdrawing it. In this suggested process, the air would be heated before each stage of expansion; and actual expansion in each cylinder would follow the intermediate practical curve between isothermal and adiabatic.

It is to be regretted that there are no indicator cards from actual expansion air-motors at hand for reproduction here, showing what results are really obtained; but these cards are singularly hard to secure. It is quite probable, however, that they would show a striking resemblance to the ordinary steam indicator card, except that their M.E.P. would be lower than that of the steam cards with the same initial back-pressure and with the same cut-off. This is explained as follows:

**Mean Effective Pressures for Air-Expansion.** In calculations for steam engines, it is generally assumed that steam expands in a cylinder in accordance with Boyle's law, or isothermally. In calculations for air-expansion, on the contrary, it is safest to assume that the expansion follows the adiabatic curve.

Thus, in Fig. 85, used by permission from Richards' "Compressed Air," a diagram is shown representing a volume of steam and the same volume of air, both at 100 lbs. gauge, and both expanded to below atmosphere. It will be noted that the steam curve is everywhere above the air curve, so that the M.E.P. of the steam during this expansion will be higher than that of the air. It is also to be observed that the air curve reaches the atmospheric line after expansion to about 4¼ times its original volume, while the steam curve reaches atmosphere after expanding 6¾ times. It thus appears that, volume for volume, at equal pressures, steam has more intrinsic power than air.

Fig. 86 shows the relations in another form. Here are a steam and an air curve, both expanding to atmosphere at the same point. They are really portions of the curves from Fig. 85, placed so that their points of intersection with the atmospheric line coincide. The air curve is now seen to be everywhere above the steam curve, showing a higher M.E.P. for the air than for the steam. But it is to be observed that while this means more power from the air, it calls for a greater initial volume of air than of steam, in the proportion of the distance *CB* to *CE*.

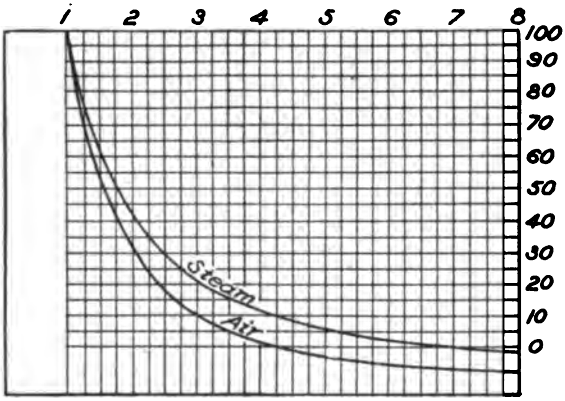


**Fig. 85. Expansion Curves for Steam and Compressed Air.**

From the foregoing, it is evident that the great distinction between an engine or motor driven by air and the same motor driven by steam, is that, the initial pressure and the required power being the same in both cases, the air-motor will require a later cut-off, and therefore a greater weight of working fluid per I.H.P., than the steam motor.

This fact will effectively condemn the use of air as against steam in all cases except where other conditions—particularly long transmissions where steam would condense with a great loss and an excessive steam consumption—justify the application of compressed air, even in the face of this disadvantage. Reheating may largely reduce the difference in favor of steam; but it can never fully wipe it out, because of the limits imposed by practical temperature conditions.

**Temperatures of Expansion.** The adiabatic compression of air from 60° F. at atmospheric pressure to 100 lbs. gauge, produces a terminal temperature of 484° F. If this air could be immediately expanded adiabatically to atmosphere, its temperature would fall to 60° F. This difference in temperature of 484 - 60, or 424 degrees corresponds to any expansion or compression with this ratio of pressures. If the air after compression is allowed to cool to initial temperature of 60° F., and then expanded adiabatically, the temperature will fall in practically the same degree—namely, 424 degrees, giving a terminal temperature at atmospheric pressure of - 364° F. If there is moisture in the air, the result in this case is very evident, even remembering that these temperatures are theoretical only and would never be actually realized. Difficulties in lubrication are also encountered in dealing with these low temperatures; and in consequence of these facts, it is very seldom attempted to use compressed air expansively in an engine or motor without reheating. The use of a reheater not only gives the added volume and improved economy already referred to, but it also gives an initial temperature before expansion begins, sufficiently high to avoid impracticably low temperatures at the motor exhaust.

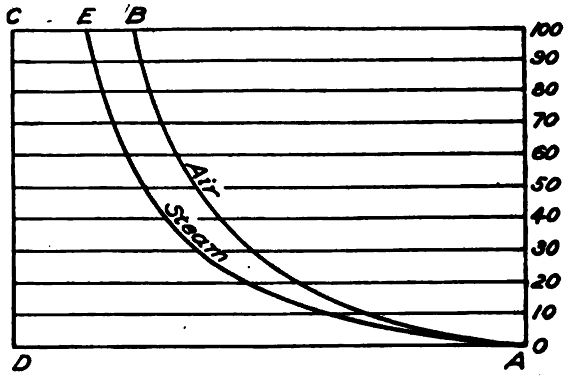


Fig. 86. Volumes of Steam and Air for Equivalent Expansions.

**Use of Compressed Air without Expansion.** While there have been successful air-power installations where the full advantages of expansion have been realized by means of reheating and careful extraction of the moisture in the air, it nevertheless remains a fact that probably 95 per cent of the applications of compressed air use the air without expansion. The greatest field of compressed-air application is that where reciprocating mechanisms are to be used, such as rock-drills, stone channelers, coal-cutting machines, pneumatic chipping and riveting hammers, etc. In all of these machines the operation is simply that of driving a piston forward and back, there being no attempt to convert this reciprocating motion into rotary motion as in the air-motor. It is practically impossible to use compressed air expansively in such machines, because there is no fly-wheel effect to come into play after cut-off. In all devices of this kind, the thing sought above all else is a heavy, powerful blow at the end of the piston stroke; and expansion would not give this desired effect. The indicator card from such machines, therefore, would be practically a rectangle with the corners rounded where the movement of the valves began a little before the end of the stroke, probably throttling slightly the intake and exhaust passages at these points.

In the small air-motors used on pneumatic drilling machines for wood and metal, there are usually several sets of cylinders arranged to give effects on the shaft equivalent to that of several cranks placed at quarters or thirds of a circle. This is not, as it might at first appear, to permit the expansive use of the air, but to give the maximum turning moment on the shaft throughout each revolution – a result which could not be secured from a single cylinder equivalent in power to all the smaller cylinders combined.

It is to be remembered that in all such machines as are here under discussion, where the air is not used expansively, the standard of economy is not in any way similar to that used for measuring the efficiency of a steam engine or of an air-driven rotary motor. The latter machines are rated in horse-powers, and their economy depends upon the steam or air consumption per horse-power-hour. But in the former machines of straight reciprocating type, the object sought is not so much a great economy of air as a great convenience and a great capacity for doing the class of work called for.

Such machines must be as light as possible, and must be capable of standing a large amount of abuse or neglect, while still doing good work. This is only one of many considerations which, in the aggregate, overbalance the importance of a high economy of air to be secured only by a greater refinement of design. With a rock-drill, for instance, the proper standard is the air consumption per foot of hole drilled; and this evidently includes many other factors than the amount of air used per stroke. With a pneumatic metal-boring drill, a proper standard should be the air consumption per cubic inch of metal drilled. Similar practical standards might be enumerated for all the special applications of compressed air.

It may be well to emphasize here the fact that the air consumption of the average compressed-air appliance, though not a thing to be ignored, can yet be much exaggerated. In the operation of a large air engine or air-driven pump working under more or less steady conditions and usually in a place where it will be looked after, carefully lubricated, packings adjusted, etc., it is well worth while to adopt all reasonable means for getting maximum power with minimum consumption of air. But with drills, coal-cutters, pneumatic hammers, and other small air-devices, which are seldom given the care and attention they deserve, the ability to "keep going" and to do good work without frequent breakdown and repairs, becomes of more importance than the air consumption. With machines of the former class, the cost of power far outweighs the cost of up-keep or maintenance; with machines of the latter class, it is not at all unusual for the repair or up-keep charges to far exceed the power charge. Questions of sturdy design and wearing power, therefore, become more important than the question of air economy.

## APPLICATIONS OF COMPRESSED AIR

It is entirely beyond the scope of this paper to deal with the details of construction of the many mechanical devices for the application of compressed air to various specific duties. Every builder of such devices has his own specialized designs, each of which is usually found to have some point of merit over others. For details of these machines, the student should refer to the catalogues of various manufacturers, the majority of which are clear and explicit, and which indeed constitute an invaluable addition to the library of the practical engineer. This paper, dealing as it does only with compressed air as a power-transmitting medium, is not concerned with the mechanical details of any air-power devices, except as they bear upon the method of control of the power which actuates them. The remainder of this paper, therefore, will contain only brief discussions of the fundamental principles of some of the more common applications of com-

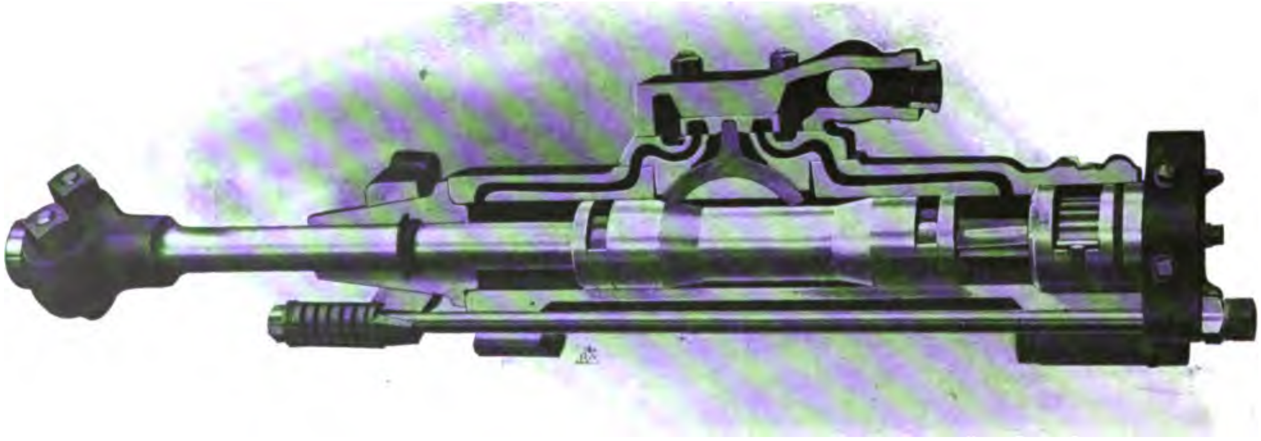


Fig. 87. Section of Tappet Valve Rock-Drill, Showing Relative Position of Working Parts.
Sullivan Machinery Co., Chicago, Ill.

pressed air. No attempt will be made to cover the entire field of compressed-air application.

**The Rock-Drill.** The rock-drill was the precursor and the cause of the air-compressor. The rock-drill was built first, and then air-compressors were designed to supply it with air. In this respect the rock-drill is the origin of all the pneumatic development of to-day.

The rock-drill is in essentials a reciprocating engine designed to give a heavy, powerful, forward stroke (the *blow*) and a very quick back stroke (the *return*). On the forward stroke, all the energy ordinarily should be applied to penetration at the drill bit, though there are certain qualities of rock which need a cushioned blow rather than a dead, stunning blow. On the return stroke, the function of which is simply to draw the steel back ready for another blow, and to *mud* the hole, a cushion must be provided to prevent the piston striking the back head of the drill. This cushion is provided by throttling the exhaust and causing compression between piston and back head, this compression helping to start the piston on its next forward stroke. But as a safeguard, in case leakage destroys this air-cushion, springs or buffers are also usually employed to absorb, by their elasticity, an accidental blow of the piston on the head. As the steel in feeding forward may sometimes encounter a hole or *pocket*, or the drill may not be fed forward rapidly enough, the front head must also be protected by some means. Frequently there is a partial air-cushion on the forward end also; but the principal reliance in this case is to be placed upon buffers or springs.

Rocks of various characters require blows of different qualities. Some are drilled most rapidly by long, swinging, powerful blows with the full stroke of the drill. Others require many short, rapid blows with only part of a full stroke. This variation in stroke is secured by manipulating the feed-screw, crowding the drill forward for short strokes, and keeping it well back for longer strokes.

Evidently this variability of stroke demands a method of valve movement equally effective on long or short piston strokes; and it entirely prevents any expansive use of the air. The rock-drill, therefore, receives and discharges practically a cylinder full of air at working pressure each stroke.

Rock-drill valve movements are of two general classes. The first form is the *tappet valve,* in which a slide-valve, balanced usually for higher pressures, is thrown by means of a tappet or rocker, which projects into the cylinder, and which is struck or moved by shoulders on the piston in its travel. This is a positive valve movement especially useful where a wide stroke variation is not necessary.

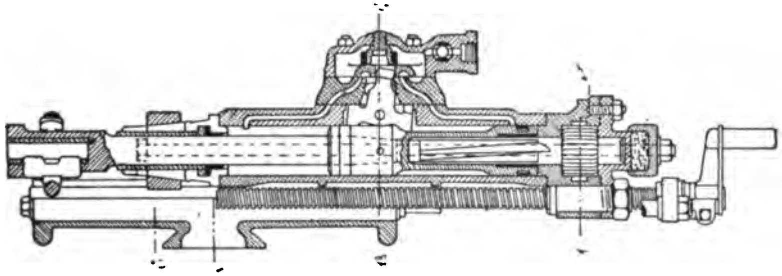


**Fig. 88. Section of "Little Giant" Tappet Valve Rock-Drill, with Balanced Valve.**
**Ingersoll Rand Co., New York City.**

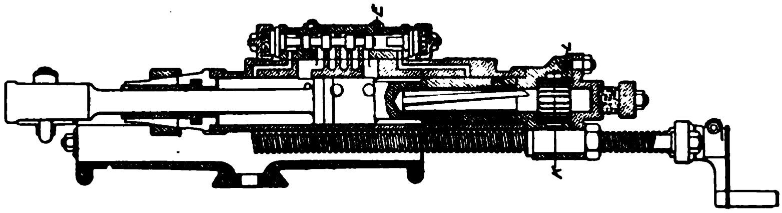


**Fig. 89. Section of "Slugger" Air-Thrown Valve Rock-Drill.**
**Ingersoll Rand Co., New York City.**

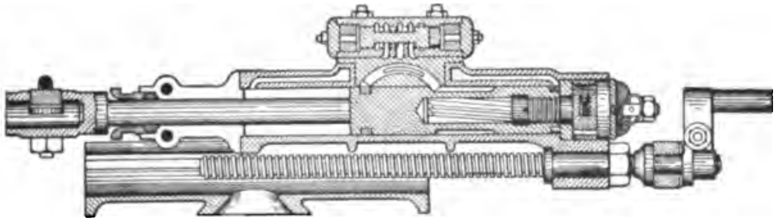


**Fig. 90. Section of "Sergeant" Rock-Drill Using a Combined Tappet and Air-Thrown Valve Movement.**
**Ingersoll Rand Co., New York City.**

Being in no way dependent for its action upon the fit of the piston in the cylinder, it will continue to operate even after wear has passed the point of good economy. The tappet action has been generally employed on drills designed to be run by steam where condensation may result in low pressures and much water in the line.

The second type of valve movement is the *air-thrown valve,* which is a spool or piston valve moved by unbalanced pressures on its two ends. A lower pressure on one end is produced by the piston uncovering in its travel a small port which exhausts part of the air in the valve-chest; full pressure on the other end of the valve then throws it, and reverses the piston movement. This valve action usually gives a more *lively* drill than the former movement, and it permits a wider variation of stroke than the tappet movement; but it is more or less dependent for its effectiveness upon the air-tight fit of all parts, and therefore suffers more under wear. This, however, is in a sense an advantage, for the action of the drill thus becomes somewhat of an index of the economy with which it is operating.

In one well-known drill on the market, these two types of valve action are combined. A tappet movement controls the admission and exhaust of air to the main valve, which in turn governs the movement of the piston. These drills combine some of the best features of both original types.

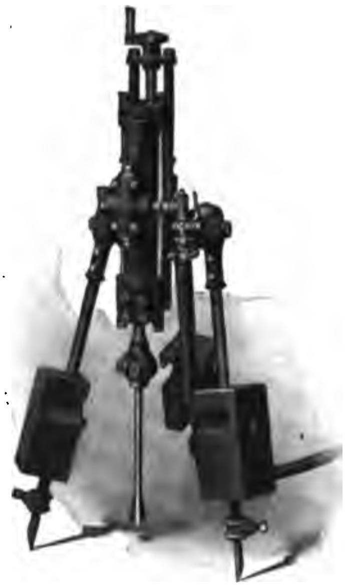


Fig. 91. Rock-Drill on Tripod.
Sullivan Machinery Co., Chicago, Ill.

**Hammer Drills.** In the ordinary rock-drill or piston drill, the cutting steel is attached to, and reciprocates with, the piston. A modern development from this scheme is the *hammer drill*, in which the steel remains stationary (except for its rotation) and is struck by the rapidly moving piston or hammer. The piston, being freed from the weight of the steel and the friction of the steel in the hole, moves much more rapidly than in the ordinary rock-drill, and the action of the hammer drill more nearly approaches that of hand-drilling. While some large hammer drills are on the market designed for the ordinary work of the piston drill, this type has found its greatest application for drilling small and comparatively shallow holes.

Hammer drills may be divided into two classes. The first has a separate valve controlling the piston, this valve usually being of the air-thrown type, but with many mechanical variations. These valve hammers have practically no cushioning, strike a very powerful blow, and make a very large number of blows per minute. The second class includes valveless hammers, in which the moving piston or hammer itself controls the admission and exhaust of air by covering and uncovering in its travel certain air-ports. The valveless hammer will probably not strike so hard a blow, nor so many of them, as a valve hammer of the same diameter; but it is often preferred because of its greater simplicity, there being but one moving part—the piston or hammer—in the machine. Choice between the

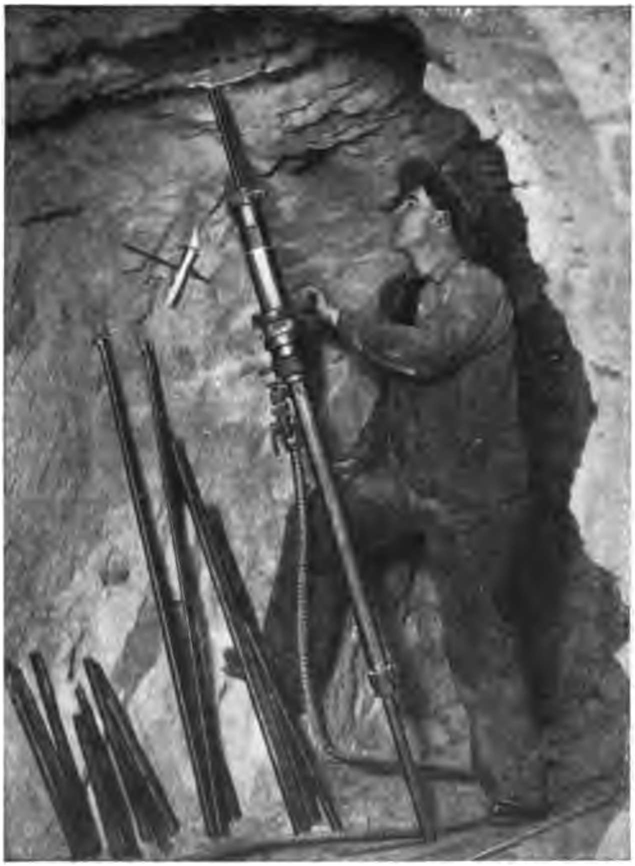


Fig. 92. Class D-21 Hammer Drill with Air Feed.
Sullivan Machinery Co., Chicago, Ill.

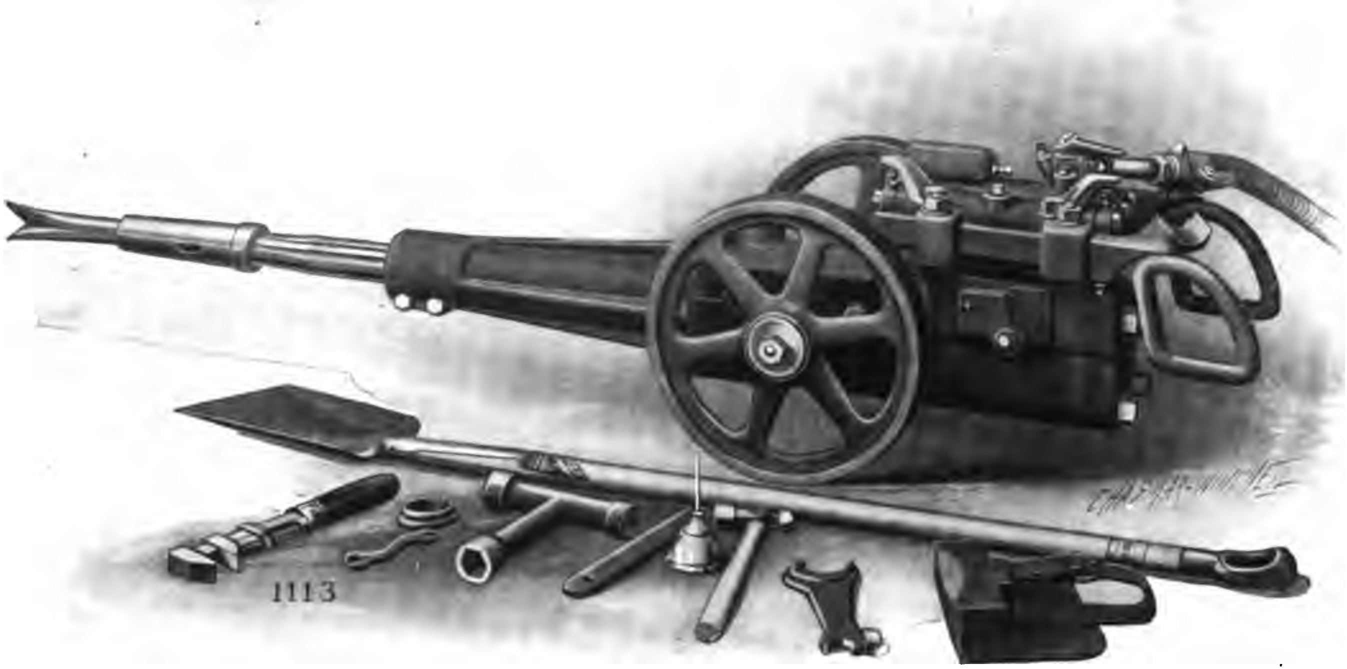


Fig. 93. "New Ingersoll" Coal Undercutter, with Complete Equipment. Ingersoll-Rand Co., New York City.

valve and valveless hammer drills seems to be largely a matter of personal preference and of local precedence in different sections of the country.

Except for the large hammer drills already mentioned as competing with the piston drills, the hammer drill is generally used without any fixed mechanical mounting. For certain classes of work, where downward holes only are required, it is fitted with a hand grip or handle, the operator pressing the tool downward against the rock; but for upward pointing holes, an air feed is used, consisting of a long piston and cylinder in which air-pressure forces the drill up against the rock. The back end of the air feed has a pointer or spud which is simply butted against the floor or wall. In certain rare classes of work, the air-feed drill is held in a mounting similar to those used with the heavier piston drills. It is seldom attempted, except in the large sizes, to provide mechanical rotation for the hammer-drill steel. Usually the steel is rotated by means of a lever or handle swung by the operator.

**Coal-Mining Machines.** The *coal-undercutting machine* or *puncher* is another reciprocating machine carrying a coal pick on its extended piston-rod. It is mounted on wheels, and is fed downward as the cut deepens, along an inclined board upon which it rolls. In this machine, cushioning is important in taking up the jar and recoil which would otherwise come upon the operator, as well as in protecting the machine itself. The valve movement may be independent, in which case, the valves will continue to keep on operating, even when the pick sticks in the coal and the piston stops; or it may be mechanically dependent upon the movement of the piston, so that when the piston stops the valve stops. The successful puncher must have a ready means of regulating the force, number, and quality of the blows it strikes, for coals of varying hardness and character. A governing device is also essential to prevent the "racing" of the machine when drawn away from the coal, to shift its position. For ordinary undercutting, the puncher is mounted on wheels of comparatively small diameter. It is sometimes used for shearing, however, or making a vertical cut in the coal face, in which case the puncher is mounted on the trunnions of wheels of large diameter, the cutting engine swinging in a vertical plane around this axis.

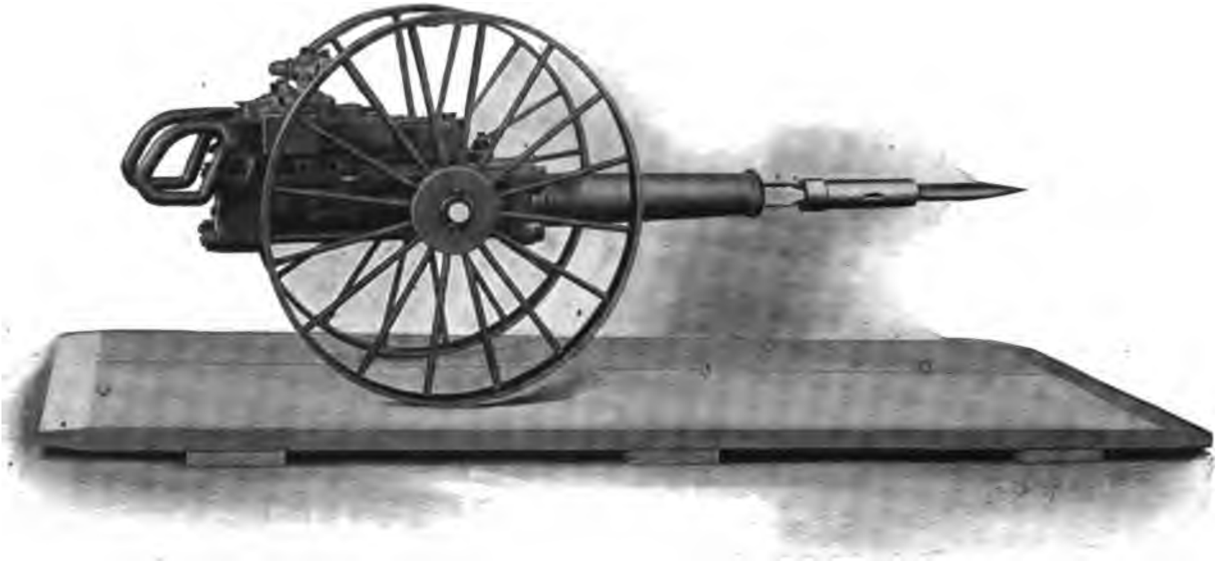


Fig. 94. Pick Machine or Puncher Mounted for Shearing. Sullivan Machinery Co., Chicago, Ill.

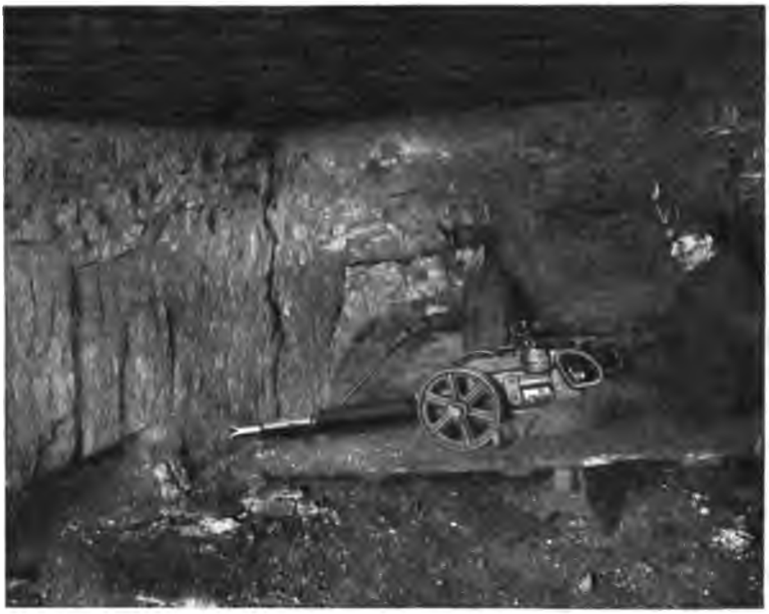


Fig. 95. Pick Machine at Ordinary Work of Undercutting in a Pennsylvania Coal Mine. Sullivan Machinery Co., Chicago, Ill.

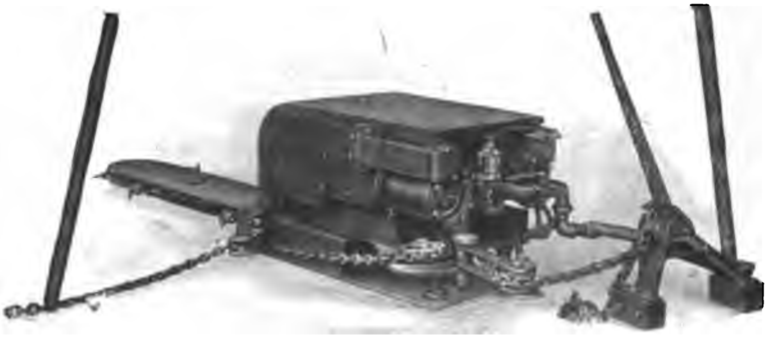


Fig. 96. Compressed-Air Chain-Type Coal-Mining Machine. Sullivan Machinery Co., Chicago, Ill.

*Coal-Shearing Machines.* A comparatively recent development in compressed-air coal-mining machines, is one for *shearing,* or making a vertical cut, in the coal vein. This device is in essentials a rock-drill with a long feed arranged on a column mounting, with a gear for swinging it in a vertical plane. Thus, as the machine is fed forward and swung around its axis, a deep, narrow, arc-shaped cut is made in the coal. This shearing cut, in combination with the undercut of the puncher, makes the shooting down of the coal easier and safer, produces a larger percentage of lump coal, and reduces the amount of powder required to bring down the face. This type of shearing machine also has a limited application as an under-cutter for use in coal veins having so steep a pitch that the ordinary puncher cannot be used. In this case the machine is arranged to swing horizontally instead of vertically.

**Track Channelers.** The track channeler is an essential in the quarrying of stone cut to exact dimensions. It consists of a heavy, reciprocating, cutting engine mounted on a truck which moves along a track, cutting a channel in the stone parallel to the track, to any depth or at any angle required. Frequently track channelers are operated by steam, each machine carrying its own boiler. But the tendency in modern quarry practice is toward the use of air-driven channelers supplied with power from a single compressing plant, this production of power in a large centralized station resulting in a greater economy of fuel and power. Not only is the cutting engine operated by air; but the engine providing the movement along the track, and frequently the motor providing for the vertical feed of the cutting steels, are also air-driven.

Channeler valve movements are always positive in character, governed by some mechanical connection with the main piston of the cutting engine. A wide variation of stroke must be provided for; and the cushioning is important because of the large diameter and heavy weight of the reciprocating parts involved. In certain qualities of rock—notably in different grades of marble—a *dead* blow shatters the material and injures it. The air-cushion in the channeler cylinder provides for the necessary variation in the character of the blow which this condition demands; and usually adjustable cushioning is provided by means of valves at both ends of the cylinder of the cutting engine.

Some modern track channelers have a *swivel-head* which permits swinging the cutting engine in the plan of the cutting steels, the maximum swing permitted being about 45 degrees either side of vertical. In addition, a *swing-back* is usually furnished, by which the cutting engine can be swung at any angle from vertical to horizontal in a plane at right angles to that permitted by the swivel-head. Such machines are useful in cutting in materials lying in diagonal strata or layers, in transferring cuts at corners, and in tunneling.

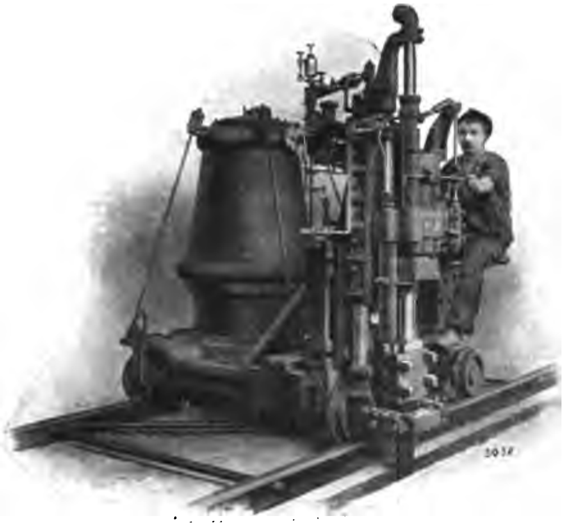
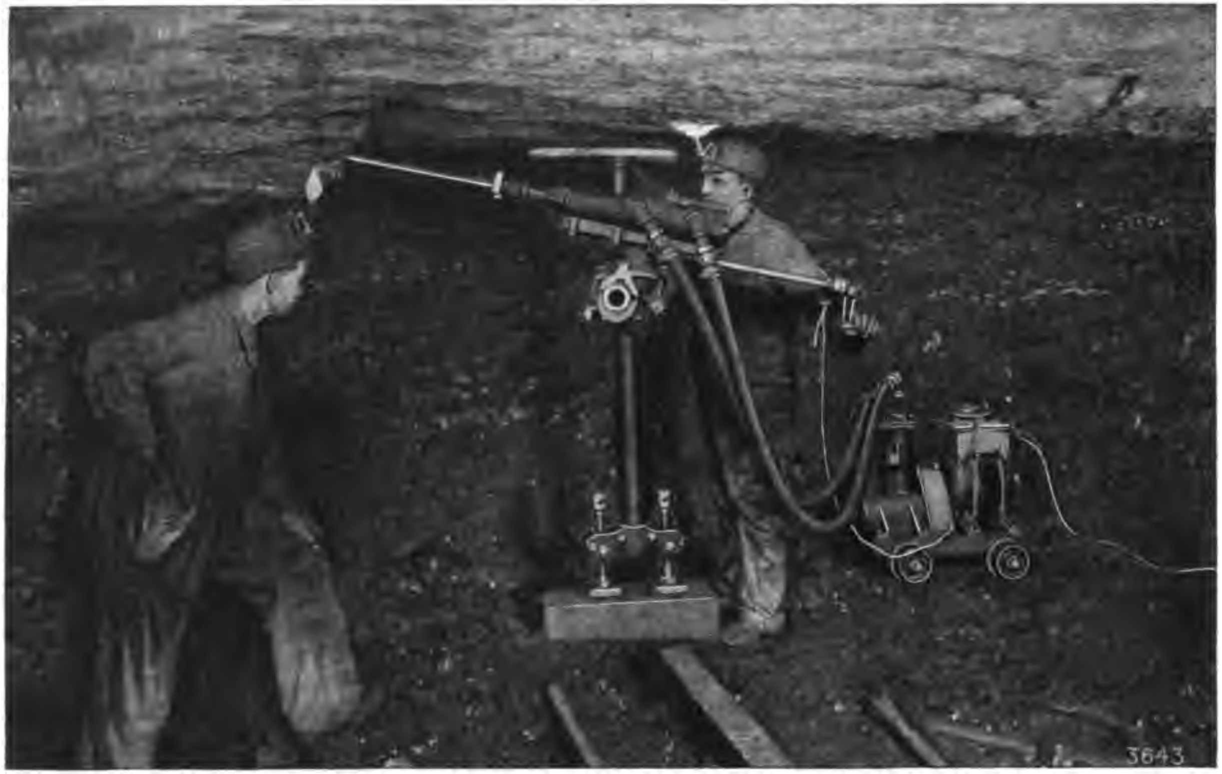


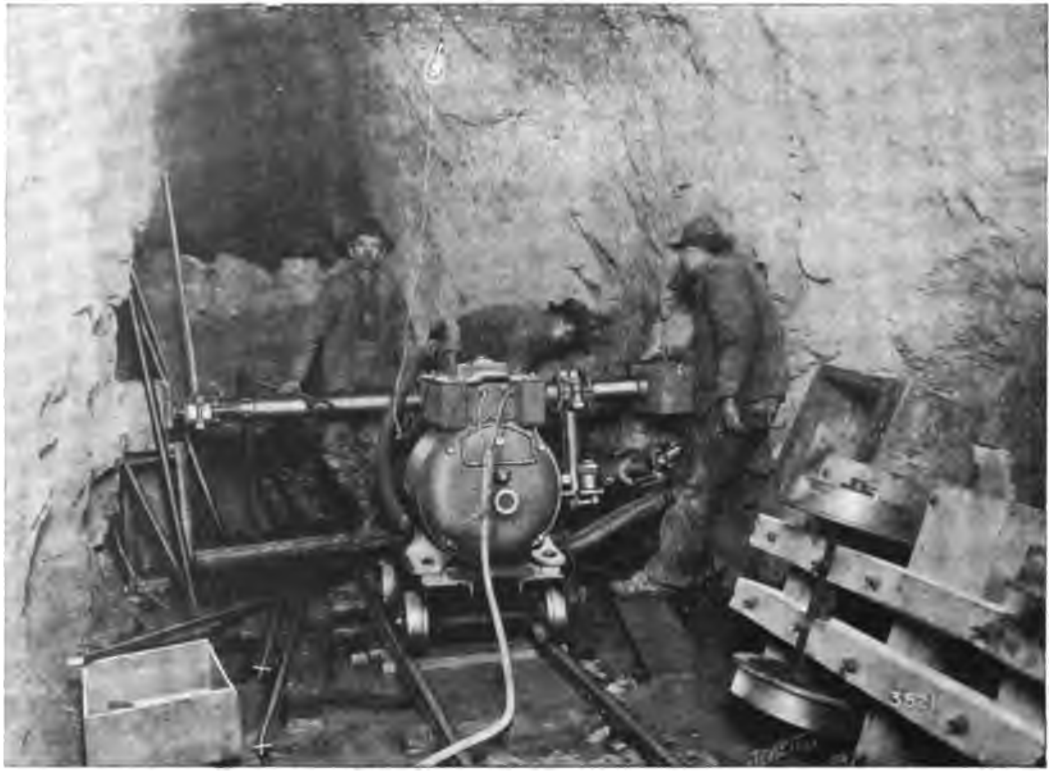
Fig. 97. H-8 "Monitor" Track Channeler, with Air-Reheater.
Ingersoll-Rand Co., New York City.

**Undercutting Track Channelers.** Where the rock which is to be quarried has no well-defined horizontal cleavage lines so that the blocks cut by the ordinary track channeler cannot be split from their bed by wedges, it is necessary to *undercut* the block. For this purpose, the *undercutting track channeler* has been designed, which is a special modification of the regular track channeler. Its cutting engine works in a horizontal plane, with an angular adjustment for corner cuts in this plane up to about 45 degrees on either side. The undercutting channeler is not so heavy a machine as the vertical track channeler, since it is never called upon for such heavy work or such deep cuts as the latter machine.

**Bar Channelers.** In general construction the cutting engine of



**AN INGERSOLL ELECTRIC-AIR DRILL BRUSHING AN ENTRY IN A COAL MINE**
A Small Motor and Air Compressor are Direct-Connected and Mounted on Trucks. In This Way, the Tool is Driven from a Compressor Nearby, and the Disadvantages of Long Distance Transmission of Air are Overcome.



AN ELECTRIC-COMPRESSED-AIR DRILL IN THE "SEVEN MILE TUNNEL," DUMONT, COLORADO
Ingersoll Sargant Drill Co.

the bar channeler is similar to that of the track channeler, but much smaller on account of its lighter duty. Instead of the whole machine moving on a track, the cutting engine of the bar channeler traverses a frame mounted on adjustable legs, movement being provided by an air-motor operating a pinion engaging a rack on the frame or bar.

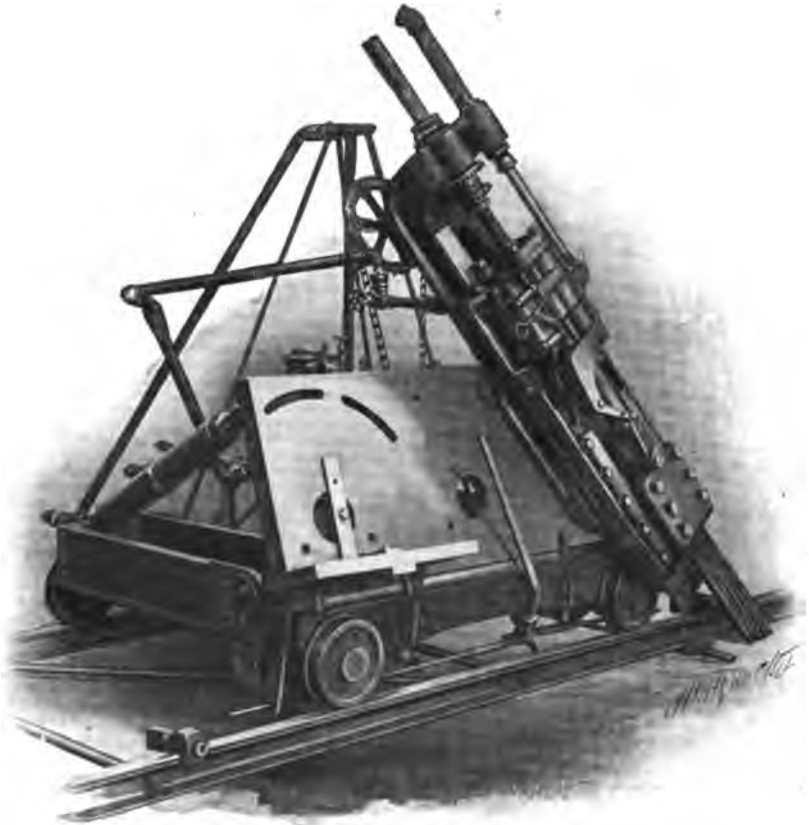


Fig. 98. Class Z Swivel-Head Swing-Back Track Channeler.
Sullivan Machinery Co., Chicago, Ill.

This type of machine is more limited in its scope than the track channeler, but has a wide range of usefulness as an accessory to the larger machine in quarries and contract work.

**Pneumatic Hammers.** Pneumatic hammers for chipping metal or stone and for driving rivets, are simply reciprocating machines in which the piston is the hammer or striking part, which travels perfectly free in the cylinder. Two general classes prevail—the *valve tool,* in which air-thrown valves in almost endless variety control the movement of the piston; and the *valveless tool,* in which the piston itself regulates the admission and exhaust of air. Very light pneumatic hammers are used for dressing and carving stone, these latter usually being known as *stone tools.* Heavier, short-stroke hammers are used for chipping and trimming metal. Long-stroke hammers prevail for riveting work. The piston or hammer of these machines is usually cushioned on air on the back stroke, to relieve the shock on the operator; but on the forward stroke a dead blow is desirable.

**Rotary Pneumatic Drills.** Rotary pneumatic drills for boring in wood or metal, for grinding and similar light work, are furnished in a great variety of sizes and patterns. But in essentials they all consist of a small air-motor—either of reciprocating or of rotary type—geared to a spindle carrying the working tool. No further description can be offered here without going into details of special constructions which would be out of place at this point.

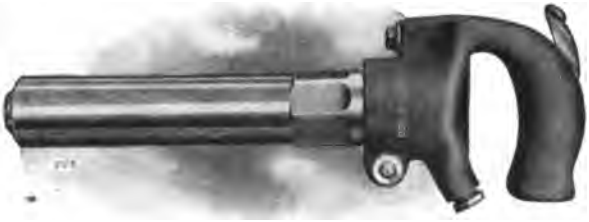


Fig. 99. "Crown" Pneumatic Chipping Hammer.
Ingersoll-Rand Co., New York City.

**Compression Riveters and Pneumatic Punches.** In these machines, air-pressure on a relatively large piston is applied, through toggles or other power-multiplying devices, to the compression of red-hot rivets, or to the punching of holes in plates or other metal forms. These are comparatively heavy machines, and their use is limited to shop and manufacturing purposes where the ready portability of the pneumatic hammer or drill is not essential.

**Pneumatic Hoists.** Pneumatic hoists are made in two general classes. The first is the *direct-lift* or *plunger hoist,* in which air

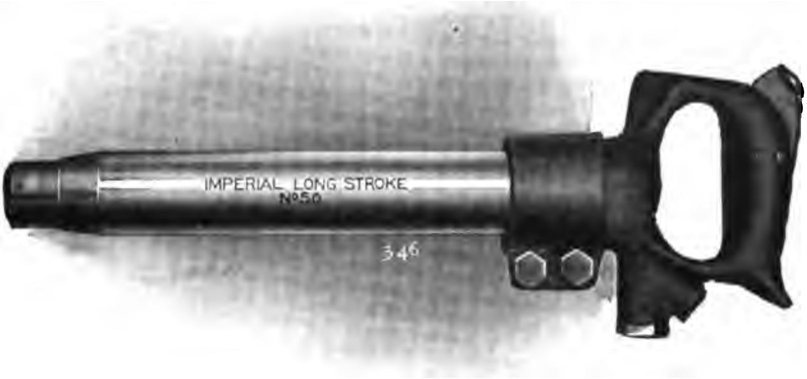


Fig. 100. "Imperial" Pneumatic Riveting Hammer.
Ingersoll-Rand Co., New York City.

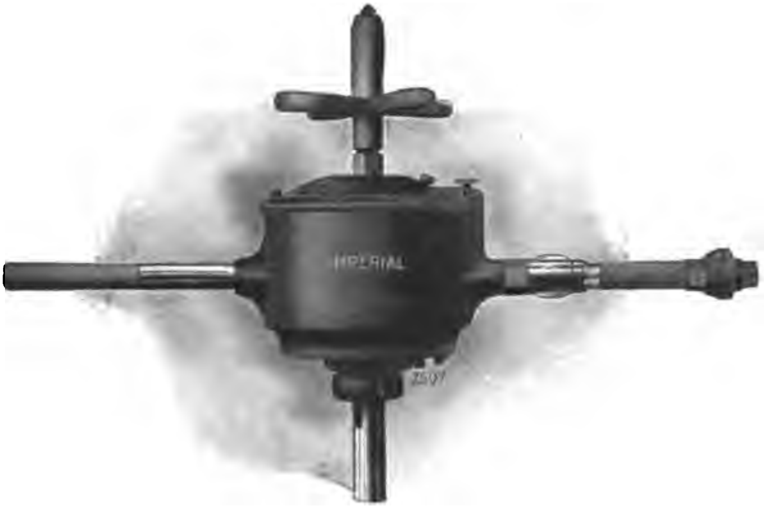


Fig. 101. "Imperial" Rotary Pneumatic Drill.
Ingersoll-Rand Co., New York City.

is admitted behind the piston of a very long cylinder, power thus applied directly lifting the weight attached to the piston-rod. These machines are necessarily limited to work where a comparatively short lift only is needed, as the longer lifts necessarily demand an inconveniently long cylinder. However, this arrangement is sometimes used in connection with cables and pulleys for providing long lifts at slow speeds, as in ice elevators and similar work. The second class is the *motor-hoist,* in which an air-motor is geared to a chain or rope drum lifting the weight. The latter machines are fully as economical and reliable as the former, and their lift is limited only by the winding capacity of their drum or sheave.

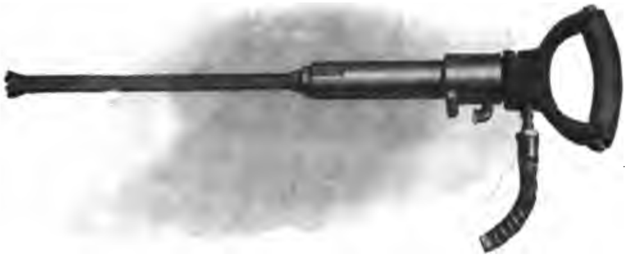


Fig. 102. Class D-19 Hand Hammer Drill.
Sullivan Machinery Co., Chicago, Ill.

**The Sand Blast.** A pneumatic sand blast for cleaning castings and building faces, for removing paint, for etching glass, etc., consists simply of a jet of air under pressure, carrying with it a stream of sharp sand directed against the surface to be operated upon. The air imparts to the sand the requisite velocity under which it simply cuts away the surface that it encounters. The air-pressure used varies among various builders of the apparatus.

**Pumping by Compressed Air.** There are three general methods of applying compressed air to the pumping of water or other fluids. In this class of work, compressed air has a distinct advantage over steam, in that there is no loss by condensation in long pipe-lines, and, though pressure may fall in transmission, there is a corresponding increase in volume, so that the net amount of energy delivered at the pump is practically the same as that put into the transmission line. The three methods of pumping by compressed air are: (1) The *direct-acting plunger pump;* (2) the *air-lift system;* and (3) the *direct-displacement pump.*

*Direct-Acting Pumps.* The ordinary direct-acting steam pump is probably the most wasteful machine in practical use to-day. It has been said of it that "it is the only method of pumping which requires as much steam (or air) to pump water down hill as to pump it up hill." The extravagance of this device is due primarily to the fact that it uses a full cylinder of steam at full pressure each stroke, with no expansion whatever. Duplex non-compounded pumps with mechanical valve movements, may use actually more than a full cylinder of steam per stroke; for the piston on one side may throw the steam valve on the other side before the piston on the latter side has finished its stroke. In such a case the live steam is discharged directly into the exhaust. All of these defects of the steam pump are found to hold when this machine is run by compressed air. It is to be noted here, however, that if an air-driven direct-acting pump is excessively extravagant of power, the fault lies in the pump itself, and not in compressed air, which stands ready to do its work economically if given a proper machine to work in.

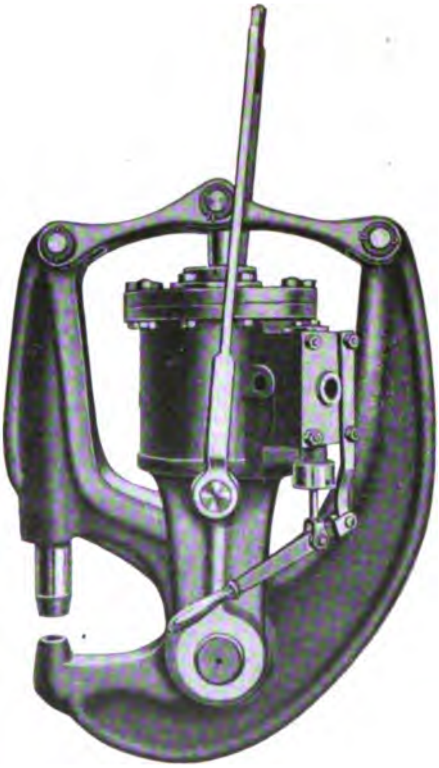


Fig. 103. Pneumatic Compression Riveter.
Chicago Pneumatic Tool Company, Chicago, Ill.

Another cause of extravagant air-consumption in the direct-acting air-driven pump, is the fact that such pumps are usually stock machines, bought without due consideration of the work they are to be used for. For instance, a pump with air and water cylinders so proportioned that air at 100 lbs. pressure will just handle a cylinder-

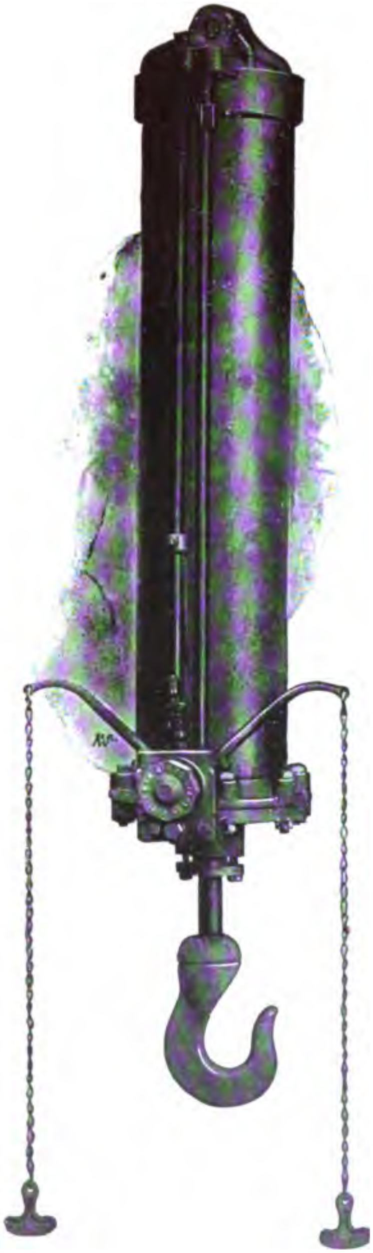


Fig. 104. Direct-Lift Pneumatic Hoist. Curtis & Co. Mfg. Co., St. Louis, Mo.

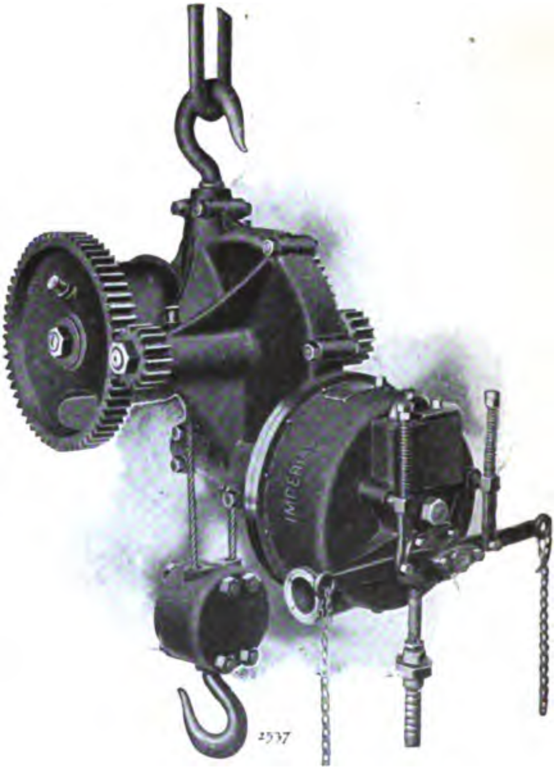


Fig. 105. "Imperial" Pneumatic Motor-Hoist, Ingersoll-Rand Co., New York City.

ful of water under a head of 200 feet, may be put at work where it pumps against a head of only 50 or 100 feet. Evidently much smaller air-cylinders could be used in the latter case than in the former; but usually the old air-cylinder is retained, using at each stroke two to four times the amount of air actually needed, simply because there is no cut-off and no means of adjusting air-consumption to the load.

Table XVI, used here by permission of the Ingersoll-Rand Company, affords the means of figuring the air-pressures and air-volumes required for a pump with the given ratios of air and water cylinders and for various lifts up to 500 feet. The method of using this table is explained by the example in the note below it.

In compound direct-acting pumps driven by air, there are four methods of utilizing the air with more or less expansion. In this connection it is to be remembered that any attempt to use compressed air expansively without reheating will result in terminal temperatures so low as to make this scheme difficult or even impossible. Some method of heating, therefore, is essential to the compounding of pump cylinders.

The first plan contemplates the heating of the air before admission to the high-pressure cylinder, to such a degree that the terminal temperature of discharge from the low-pressure cylinder is within a practical limit. This is open to the objection that so high an initial temperature would be required that proper lubrication of the high-pressure cylinder would be almost impossible. Moreover, since there is no fly-wheel on the pump, cut-off in either cylinder, with the subsequent expansion, would mean a reduction in power at the end of the stroke, and a loss of speed, so that the movement would be irregular.

The second plan utilizes both preheating before high-pressure admission and interheating between the cylinders. While the heating between cylinders permits a reduction of the initial temperature over that required by the first method, the second objection cited in the first method would still hold.

The third method uses air without heating and without cut-off or expansion in the high-pressure cylinder; but this air is expanded after high-pressure exhaust into the low-pressure cylinder, where it is used either expansively or with no cut-off. Unless interheating is used, a troublesome drop in temperature will probably result from

## TABLE XVI

### Compressed-Air Table for Pumping Plants

Reasonable allowances have been made for loss due to clearances in pump and to friction in pipe.

| RATIO OF DIAMETERS | PERPENDICULAR HEIGHT, IN FEET, TO WHICH THE WATER IS TO BE PUMPED | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 25 | 50 | 75 | 100 | 125 | 150 | 175 | 200 | 225 | 250 | 300 | 350 | 400 | 450 | 500 | |
| 1 to 1 | 13.75 | 27.5 | 41.25 | 55.0 | 68.25 | 82.5 | 96.25 | 110.0 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | Air-pressure at pump |
| | 0.21 | 0.45 | 0.60 | 0.75 | 0.89 | 1.04 | 1.20 | 1.34 | ..... | ..... | ..... | ..... | ..... | ..... | ..... | Cubic feet of free air per gallon of water |
| 1½ to 1 | ..... | 12.22 | 18.33 | 24.44 | 30.33 | 36.66 | 42.76 | 48.88 | 55.0 | 61.11 | 73.32 | 85.4 | 97.66 | ..... | ..... | Air-pressure at pump |
| | ..... | 0.65 | 0.80 | 0.95 | 1.09 | 1.24 | 1.39 | 1.53 | 1.68 | 1.83 | 2.12 | 2.41 | 2.70 | ..... | ..... | Cubic feet of free air per gallon of water |
| 1¾ to 1 | ..... | ..... | 13.75 | 19.8 | 22.8 | 27.5 | 32.1 | 36.66 | 41.25 | 45.83 | 55.0 | 64.16 | 73.33 | 82.5 | ..... | Air-pressure at pump |
| | ..... | ..... | 0.94 | 1.14 | 1.24 | 1.30 | 1.54 | 1.69 | 1.84 | 1.99 | 2.39 | 2.59 | 2.88 | 3.19 | ..... | Cubic feet of free air per gallon of water |
| 2 to 1 | ..... | ..... | ..... | 13.75 | 17.19 | 20.63 | 24.06 | 27.5 | 30.94 | 34.38 | 41.25 | 48.13 | 55.0 | 61.88 | 68.75 | Air-pressure at pump |
| | ..... | ..... | ..... | 1.23 | 1.37 | 1.52 | 1.66 | 1.81 | 1.96 | 2.11 | 2.40 | 2.69 | 2.98 | 3.28 | 3.57 | Cubic feet of free air per gallon of water |
| 2¼ to 1 | ..... | ..... | ..... | ..... | 13.75 | 16.5 | 19.25 | 22.0 | 24.75 | 27.5 | 33.0 | 38.5 | 44.0 | 49.5 | 55.0 | Air-pressure at pump |
| | .... | ..... | ..... | ..... | 1.533 | 1.68 | 1.83 | 1.97 | 2.12 | 2.26 | 2.56 | 2.85 | 3.15 | 3.44 | 3.73 | Cubic feet of free air per gallon of water |
| 2½ to 1 | ..... | ..... | .... | ..... | ..... | 13.2 | 15.4 | 17.6 | 19.8 | 22.0 | 26.4 | 30.8 | 35.2 | 39.6 | 44.0 | Air-pressure at pump |
| | .... | ..... | ..... | .... | ..... | 1.79 | 1.98 | 2.06 | 2.104 | 2.34 | 2.62 | 2.88 | 3.18 | 3.36 | 3.23 | Cubic feet of free air per gallon of water |

**To find the amount of air and pressure required to pump a given quantity of water a given height, find the ratio of diameters between water and air cylinders, and multiply the number of gallons of water by the figure found in the column for the required lift. The result is the number of cubic feet of free air. The pressure required on the pump will be found directly above, in the same column. For example: The ratio between cylinders being 2 to 1, required to pump 100 gallons, height of lift 250 feet. We find under 250 at ratio 2 to 1, the figures 2.11: 2.11 × 100 = 211 cubic feet of free air. The pressure required is 34.38 pounds delivered at the pump piston.**

the expansion between cylinders — a difficulty further increased where expansion in the low-pressure cylinder is attempted.

The fourth method uses the air at full stroke in both cylinders; but between the cylinders the exhaust from the high-pressure cylinder is reheated to a point where its original pressure is restored. By this method there is no expansion or drop in temperature in the cylinders, and speed is uniform. This method has been successfully applied in triple-expansion pumps using three air cylinders with two interheaters, giving a very high economy.

To get the best economy (never very good, however) from the simple direct-acting air-driven pump, the clearances should be made as small as possible and the cylinder ratio made right for the work in hand. From air-driven compound direct-acting pumps, no great degree of success can be expected unless one of the four methods above described is used; and the fourth method gives the best results.

The mechanical features of direct-acting air-driven pumps are similar to those of direct-acting steam pumps.

*Air-Lift Pumping System.* The air-lift was originally designed for deep-well pumping, and still finds its greatest application in that field; but it has also been successfully applied for mine pumping and for industrial work, particularly in handling solutions of varying chemical possibilities. A small air-pipe is carried down the well, and compressed air is discharged in the water discharge or *eduction* pipe at the bottom. This air, mingling with the water in the eduction pipe, gives this column of mixed air and water a lower specific gravity than that of the solid water column outside the pipe. There is, therefore, an unbalanced condition of affairs; and the greater weight of the outer mass of water forces the column of mingled air and water in the pipe upward and out. This is the principle of the air-lift in its simplest terms.

There are no generally accepted rules for designing air-lift systems. The question is one involving the depth of water below the surface, the *submergence* or length of eduction pipe below the water level during pumping, the total lift or distance water must be elevated, the diameter of the eduction pipe, the volume and pressure of the air-supply, etc. Different companies furnishing this system appear to have different rules for manipulating these several factors. Various *foot-pieces* are also supplied by different builders, for securing the proper mingling of the air and water at the bottom of the eduction pipe.

A properly designed air-lift system will probably pump from a driven well at higher efficiency than any other method of pumping, particularly if the cost of up-keep, as well as of power, is taken into account. The ordinary deep-well pump has most of its working mechanism at the bottom of the well, submerged, inaccessible, and subject to the damaging action of sand, dirt, and grit. The air-lift, on the contrary, has nothing in the well to wear out, nothing but some pipes exposed to the action of the fluid; and all its working mechanism is the compressor, always under the eye of the operating engineer and presumably kept at highest efficiency. One compressor plant, moreover, can pump any number of wells scattered over the water-bearing area; and any or all of these wells can be instantly controlled from the compressor plant. The air-lift system is distinguished by a very low cost for wells and equipments. All of these advantages are equally applicable where the system is used for other purposes than deep-well pumping. The fact that this device has no working parts exposed to the fluid pumped, is of particular value where acid or corrosive solutions are to be handled.

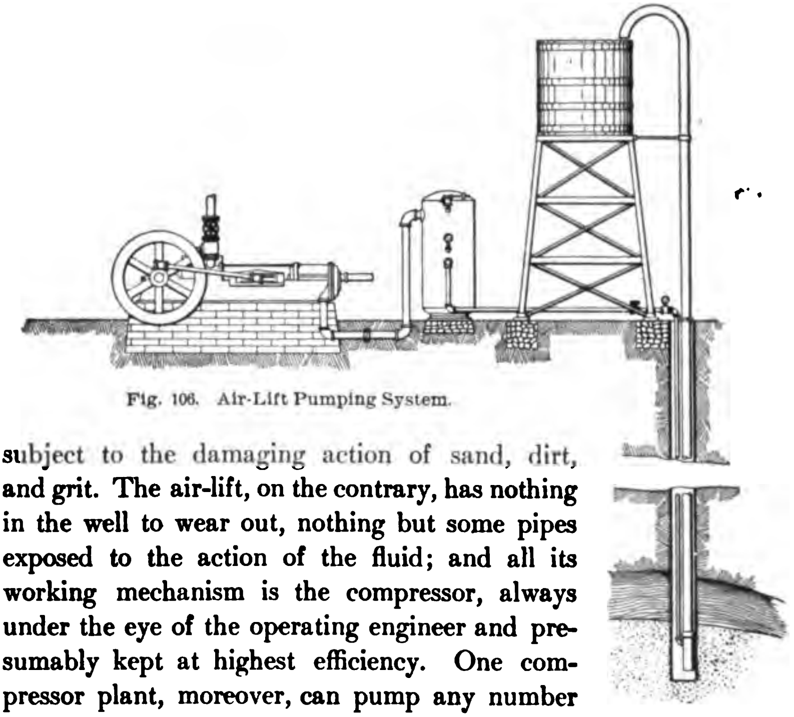


Fig. 106. Air-Lift Pumping System.

*Direct-Displacement Pump.* The pneumatic direct-displacement pump consists of one or more tanks submerged in the fluid to be pumped, so that they are filled by gravity. When a tank is filled with the fluid, air under suitable pressure is admitted at the top of the tank, forcing out or displacing the volume of liquid. As soon as the liquid has been displaced, the air-volume is discharged or exhausted to atmosphere, and the tank again fills by gravity and is again discharged under air-pressure. This process continues automatically as long as the pressure is supplied. Usually the tanks are arranged in pairs, so that while one is being discharged the other is being filled. Alternate admission and discharge of compressed air are controlled by valves operated by floats or buckets in the tanks, so disposed that as soon as a tank is completely filled air is admitted, and the air-admission closed as soon as the tank is emptied. The volume discharged is limited only by the size of the tanks, and the height of the lift is limited only by the air-pressure available.

The displacement pump, even at its best, is open to two objections, which, however, for limited lifts, are often more than offset by the perfect simplicity and sturdy reliability of the device. The first objection is that the valve mechanism must be on the tanks, and therefore submerged with them. This means that this mechanism is inaccessible for examination and adjustment, and is fully exposed to the action of the fluid pumped. In some of the better constructions, however, the valve movement is enclosed so that the latter difficulty is minimized. The second objection is that the discharge of a tank full of air at full pressure is a great waste of power, and evidently this objection increases as the lift, and therefore the pressure required, increases. However, as stated, the direct-displacement pump has distinct advantages often recommending it for use, even in spite of these shortcomings.

*Return-Air Pumping System.* One of the large builders of compressed-air machinery has produced a modified direct-displacement pumping system designed to obviate the two objections just cited in the case of the plain displacement pump. This system is known as the *return-air system,* and is an application of the closed-pipe or dense-air system of power transmission mentioned earlier in this paper. With this device, twin tanks are used; but the valve mechanism governing the admission and discharge of air, instead of being mounted on the tanks, is located in the compressor room. The air, after displacing the fluid, is not discharged to atmosphere, but is carried back through a return pipe to the intake side of the compressor. Thus its energy of expansion is applied on one side of the compressor piston, helping to do the work of compression on the other side. In this arrangement a single-stage compressor is ordinarily used, since the ratio of compression in the cylinder is small, though the terminal pressure in the machine itself may be relatively high. This system, therefore, removes the valve mechanism from the action of the fluid pumped, and places it in the compressor room where it can be kept in proper adjustment. The air is also used with higher economy. There is, of course, some leakage of compressed air in the system; and this is compensated by a check-valve on the piping, which opens to admit free air when pressure falls below a certain point.

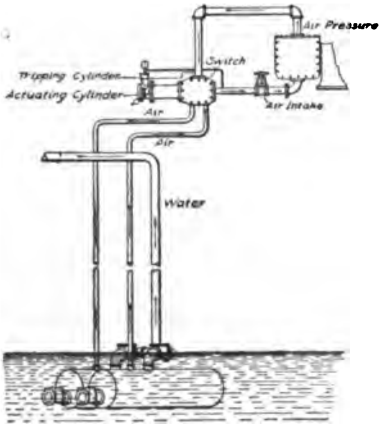


Fig. 107. Diagram of Return-Air Pumping System.
Ingersoll-Rand Co., New York City.

The manufacturers of the return-air system also offer the direct-displacement pump, and in their literature fix the distinction between the proper applications of these two systems as follows: The plain direct-displacement pump is recommended for lifts up to 50 feet; for lifts of 50 feet to 250 feet, the plain return-air system is recommended; and for lifts of above 250 feet, a compounded return-air system is offered.

**The Electric-Air Principle.** Another example of the application of the dense-air system is found in the *electric-air drill* and *electric-air channeler* produced by a prominent builder of pneumatic machinery. In these machines each device consists of a small, complete air compressing, transmitting, and applying system. Air is compressed by a small motor-driven duplex air-compressor designated as the *pulsator*. From each of the pulsator cylinders, a length of hose or pipe leads to one end of the cylinder of the reciprocating engine of the drill or channeler. The air, however, is not compressed and discharged to atmosphere, but is compressed and expanded back into the pulsator, where it is recompressed and re-expanded; and so on, indefinitely. Air compressed in one pulsator cylinder passes through a length of hose to one end of the reciprocating engine, driving the piston in the latter forward. At the same time, the air on the other side of the reciprocating piston is withdrawn from the cylinder through the other length of hose and into the other pulsator cylinder. On the return stroke, this process is reversed. Thus there is had in effect a compressor and a reciprocating engine without any inlet or discharge valves, the air being simply forced back and forth under pressure in a closed circuit. The full advantage of the energy of expansion contained in the air at the end of the stroke, is secured; and, by making the piston of the reciprocating engine relatively large, a powerful blow is secured without the use of high air-pressures. The alternate compression and expansion of the air in the pulsator evidently makes unnecessary the use of any water jacket. These machines are put out by the builder under a guarantee that they will do the work of an

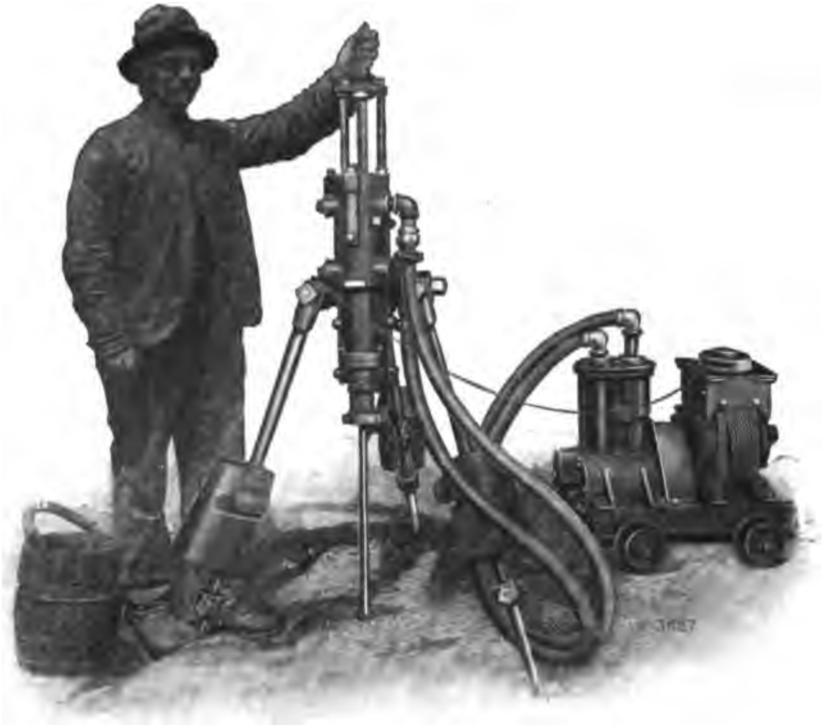


Fig. 108. "Electric-Air" Rock-Drill with Alternating-Current Pulsator.
Ingersoll-Rand Co., New York City.

equivalent machine of the ordinary pattern, with a saving of at least one-half the power. This higher economy is due to the use of the lower pressures, and to the saving by re-expansion of the compressed air.

**Electro-Pneumatic Switch and Signal Systems.** In the electro-pneumatic switch-and-signal system for railway service, compressed air from some central plant furnishes the power for throwing the switches and operating the signals, the movement being produced by the action of air on the piston of a cylinder. Electricity in this system is simply the controlling force used for manipulating the valves for admitting and discharging the air operating the heavier mechanism. This subject is a large one, and no attempt will be made at this point to go into the details of construction and operation.

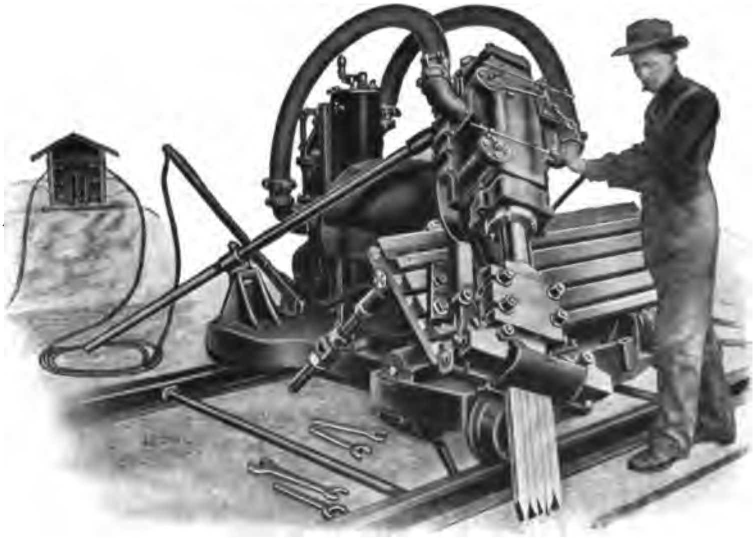
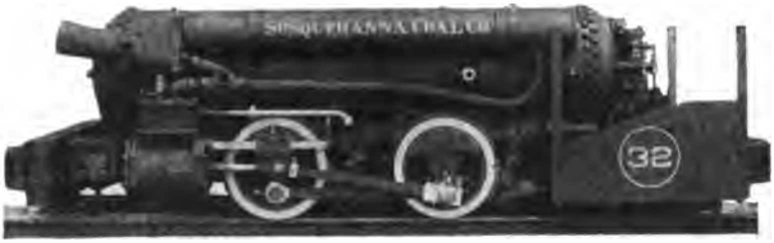
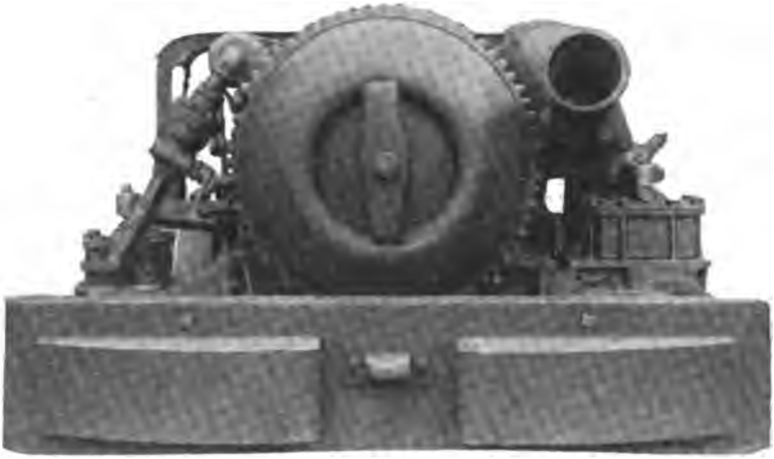


Fig. 109. "Electric-Air" Track Channeler.
Ingersoll-Rand Co., New York City.

**Pneumatic Haulage.** Locomotives operated by compressed air are largely used for haulage purposes in mining and industrial work. These machines use air at high pressure, carried in one or more storage tanks, and delivered to the cylinders through a reducing valve so that the actual working pressure is moderate. For haulage work, the air is compressed in multi-stage high-pressure machines, and delivered in pipes to various charging stations along the haulage track. The storage pressure is usually from 900 to 1,200 lbs., which is reduced to from 100 to 150 lbs. as delivered to the cylinders. The reducing valve automatically maintains a constant delivery pressure, whatever the pressure in the storage tanks, the latter of course falling as air is withdrawn.



Side View.



End View.

Fig. 110. Side View, and End View (at larger scale), of Compressed-Air Locomotive. Compound type. The air, as it leaves the high-pressure cylinder at a very low temperature, passes through an atmospheric interheater before reaching the low-pressure cylinder. By actual tests, this gives an increase in efficiency over the simple-acting locomotive of 50-65 per cent.
H. K. Porter Co., Pittsburg, Pa.

In small locomotives, the air is ordinarily used in the cylinder without cut-off and expansion. This of course reduces the economy, but results in giving large power from a very compact machine with small cylinders. In locomotives of large size, there is usually provision for adjustable cut-off on the cylinders. Thus no cut-off can be used in starting a train or going over a heavy grade; and the cut-off is shortened when the load is light or the grade easy, when less power is required. It is true that this expansive use of the air produces a low temperature. But the initial storage pressure of the air is so high that, when this air is expanded through the reducing valve and after cut-off, the percentage of moisture in the air is so small that no trouble is caused by freezing. Nevertheless, large, short, direct ports and passages are usually sought in the design. Reheating is sometimes employed, to improve still further the economy. The locomotive cylinders are not covered, but left open to the air so that as much heat as possible may be absorbed from the atmosphere. In some cases the exterior of the cylinder is ribbed, to expose a larger surface for heat absorption.

## COMPRESSED-AIR BIBLIOGRAPHY

The following publications are recommended to the student wishing to pursue further the study of compressed air:

*Cyclopedia of Compressed Air Information*, edited by W. L. Saunders; Compressed Air Magazine Co., New York City, Publishers.

*Compressed Air in All Its Applications*, by Gardner D. Hiscox; Norman W. Henley Publishing Co., New York City, Publishers.

*Compressed Air*, by Frank Richards; John Wiley & Sons, New York City, Publishers.

*Compressed Air Plant for Mines*, by Robert Peele; John Wiley & Sons, New York City, Publishers.

*Mechanics of Air Machinery*, by Weisbach and Herman; D. Van Nostrand Co., New York City, Publishers.

In addition to these books, the student is referred to *Compressed Air Magazine* (monthly), published by the Compressed Air Magazine Company, New York City.

# INDEX

A

Page

- Absolute pressure, definition of . . . . 2
- Absolute temperature, definition of . . . . 2
- Accumulator . . . . 118
- Adiabatic compression . . . . 7
- Adiabatic expansion . . . . 129
- Aftercooler . . . . 117
- Air
  - composition of . . . . 2
  - physical properties of . . . . 3
  - power required to compress . . . . 27
- Air-compression
  - compound or stage . . . . 81
  - at different altitudes . . . . 28
  - ideal . . . . 9
  - unavoidable losses in . . . . 31
  - water-jacketing . . . . 79
- Air-compressor valves . . . . 91
  - air-discharge . . . . 98
  - air-inlet . . . . 96
  - inlet . . . . 91
  - mechanically actuated . . . . 92
  - poppet . . . . 91
- Air-compressors
  - compound steam cylinders on . . . . 43
  - duplex . . . . 67
  - high-pressure . . . . 103
  - ignition and explosions . . . . 101
  - mechanical efficiency of . . . . 100
  - straight-line . . . . 60
  - vertical . . . . 60
- Air-discharge valves . . . . 98
- Air-expansion, mean effective pressures for . . . . 131
- Air-inlet valves . . . . 96
- Air-lift pumping system . . . . 155
- Air-motor indicator cards . . . . 129
- Air-power plant, design of . . . . 106
- Air-power transmission, tables of . . . . 110
- Air-receivers
  - primary . . . . 118
  - secondary . . . . 120

Page
Air-thrown poppet valve........ 92
Air-thrown valve........ 138
Antecooler........ 116
Applications of compressed air........129, 135
Atmospheric pressure, definition of........ 2

B

Balance wheel of air-power system........ 119
Bar channelers........ 146
Belt drive........ 50
Belt-driven compressor........ 33
Bibliography, compressed-air........ 162
Boyle's law........ 4

C

Chain-driven compressors........ 33
Channelers
  bar........ 146
  electric-air........ 158
  track........ 145
Charles's law........ 5
Clearance space, effect of........ 17
Coal-mining machines........ 142
Coal-shearing machines........ 145
Composition of air........ 2
Compound air-compression........ 81
  advantages of........ 87
  intercooler........ 85
Compounding, steam pressures for........ 44
Compressed air........1–162
  application of........129, 135
  bibliography........ 162
  compressors........ 32
  indicator card........ 11
  production of........ 1
  pumping by........ 150
  reheating........ 124
  transmission and distribution of........ 109
  use of, without expansion........ 133
  valves........ 91
  water jackets........ 79
Compression
  adiabatic........ 7
  heating effect of........ 10
  isothermal........ 5
  loss of work due to heat of........ 11
  necessity for cooling during........ 8
  phenomena of........ 6

Page

Compression
    pressures and volumes in........ 7
    removal of moisture during........ 117
Compression efficiency, determining........ 18
Compression riveters........ 148
Compressors
    classification of........ 32
    power-driven........ 50
    steam-driven........ 43
Curves, laying out isothermal and adiabatic........ 18

D

Dense-air system of distribution........ 122
Direct-acting pumps........ 151
Direct-displacement pump........ 156
Distribution
    dense-air system........ 122
    loop system........ 121
Dry air, necessity for........ 116
Duplex air-compressor........ 33, 67
    advantages of........ 74
    pressure relations in........ 69

E

Electric-air channeler........ 158
Electric-air drill........ 158
Electro-pneumatic switch and signal systems........ 160
Expansion, temperatures of........ 133

F

Free air, definition of........ 1

G

Gas-compressors........ 106
Gear drive........ 52
Gear-driven compressors........ 33
Governors........ 46

H

Hammer drills........ 139
Heating effect of compression........ 10
High-pressure air-compressors........ 103
Horizontal compressors........ 33
Horizontal receiver........ 119

I

Ideal air-compression........ 9
Indicator card
    compressed-air........ 11

Page

Indicator card
  from multi-stage compressors . . . . . 23
  representative . . . . . 23
Inlet and discharge port areas . . . . . 99
Inlet valves . . . . . 91
Intercooler . . . . . 85
Internal-combustion reheaters . . . . . 126
Isothermal compression . . . . . 5
Isothermal expansion . . . . . 129

L

Loop system of distribution . . . . . 121
Loss of work due to heat of compression . . . . . 11

M

Mean effective pressures for air-expansion . . . . . 131
Mechanical efficiency of air-compressors . . . . . 100
Mechanical mixture . . . . . 2
Mechanically actuated valves . . . . . 92
Moisture, removal of, during compression . . . . . 117

N

Nitrogen . . . . . 2

O

Oxygen . . . . . 2

P

Perfect gas, definition of . . . . . 3
Phenomena of compression . . . . . 6
Physical properties of air . . . . . 3
Pipe-line . . . . . 120
Pneumatic hammers . . . . . 147
Pneumatic haulage . . . . . 160
Pneumatic hoists . . . . . 148
Pneumatic punchers . . . . . 148
Power-driven compressors . . . . . 32, 50
  belt drive . . . . . 50
  gear drive . . . . . 52
  regulation of . . . . . 53
  rope drive . . . . . 51
  starting unloaders for . . . . . 59
Power required to compress air . . . . . 27
Pressures and volumes in compression . . . . . 7
Pressures, volumes, and temperatures, relation of . . . . . 20
Primary air-receivers . . . . . 118
Production of compressed air . . . . . 1
Pulsator . . . . . 158
Pumping by compressed air . . . . . 150

Page

Pumping by compressed air
  air-lift pumping system.... 155
  direct-acting pumps.... 151
  return-air pumping system.... 157

R

Rectifier.... 118
Regulation of power-driven compressors.... 53
Regulation of steam-driven compressors.... 46
Reheating compressed air.... 124
Return-air pumping system.... 157
Rock-drill.... 137
Rope drive.... 51
Rope-driven compressor.... 33
Rotary pneumatic drills.... 148

S

Sand blast.... 150
Silent-chain drive.... 52
Single-stage compressor.... 33
Speed-limiting device for governors.... 48
Steam-driven compressors.... 32, 43
  regulation of.... 46
Steam-engine and compressor practice, distinction between.... 43
Steam pressures for compounding.... 44
Straight-line air-compressor.... 33, 60
  pressure relations in.... 63

T

Table
  air, H. P. developed in compressing one cu. ft. of free, etc.... 27
  air, H. P. required to compress, etc.... 30
  air-compression at different altitudes.... 28
  air-compression at various altitudes, relation of volumetric efficiencies and H. P. required in.... 31
  compressed-air table for pumping plants.... 154
  compressing air from one atmosphere and 60° F., volumes, pressures, and temperatures, etc., in.... 21
  compression, efficiency of, etc.... 17
  dry air, weights and volumes of, at atmospheric pressure, etc.... 4
  globe valves, tees, and elbows.... 115
  heat of compression, loss of work due to, etc.... 11
  isothermal and adiabatic pressure, data.... 20
  loss of pressure in lbs., by friction, in transmission of air pipes, 1,000 ft. long.... 111–114
  temperatures due to adiabatic compression, etc.... 10
Tappet valve.... 138
Temperatures of expansion.... 133
Thermodynamics.... 6

Page
Track channelers.......... 145
Transmission and distribution of compressed air.......... 109

U

Undercutting track channelers.......... 146
Unloaders for power-driven machines, starting.......... 59

V

Valves, ports, and passages, areas of.......... 99
Valves, tees, and elbows.......... 115
Vertical compressors.......... 33, 60
Vertical receiver.......... 119

W

Water-jacketing.......... 79

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